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GAS-ENGINE DESIGN

BY

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PREFACE.

The purpose of this book is to present in a compact form those principles which underlie the design of gas-engines, together with such data on the subject as seem reliable for the use of those engaged in building this kind of machinery, and who are familiar with its characteristics. The qualitative or inventive side of design, such as is treated in all the books that have so far appeared, except Güldner in German, is here entirely omitted and familiarity with such presupposed. This book is concerned entirely with the quantitative side of design, and treats solely of the forces in, and the energy-transforming power of the standard mechanism of, the exploding gas-engine.

All those whose interests have demanded such a quantitative knowledge of the gas-engine, either for probable output and economy or for the stresses in and proper strength of resisting engine parts, have met with difficulty in finding reliable data for reference, as there is no book in English treating exclusively of this side of the subject. The data here presented are the result of many years' collection and personal experience, and were first classified in the present form for lecture use before my classes at Columbia University. The increase in quantity of material during the last few years made it seem desirable to publish the notes in as closely condensed a form as possible consistent with clearness.

The work is divided into three parts. The first, treating of power, efficiency, and economy, gives the material necessary for deciding on the necessary piston displacement for any specified output for any kind of gas, and enables the designer to approximately predict economy. The second part contains the data and method for determining the stresses in the parts and the number and arrangement of cylinders necessary for balance or turning effort to meet the specifications. The last part is entirely concerned with the dimensions of the parts to resist the stresses, both by theoretic analysis and by empirical formulæ, showing between what limits every principal dimension should lie.

These results apply to all classes of gas-engines, from the small high and variable-speed automobile and boat engine to the large 5000-H.P. constant-speed station engine.

The method of treatment used in the first part is believed to be new. A standard reference indicator card is established and, by comparison with tests, a diagram factor found by which the probable M.E.P. for any fuel and compression can be predicted. It is further shown that this same diagram factor, applied to the efficiency of the standard card, will give the probable efficiency of the engine as well as the economy when applied to the standard economy. The diagram factor can therefore be used for three purposes, making calculations easy and certain; quite contrary to steam work, where the M.E.P. diagram factor is very different from the ratio of theoretic to actual water-rate, or efficiency. In the second part means are given for drawing the probable indicator card and, by tables and curves, the probable inertia diagram for any engine yet unbuilt. From these the stresses are found, as well as the turning effort, for all conditions of load, ignition, charge, mixture, speed, weight of parts, and combinations of cylinders. Means are also presented here for quickly estimating the conditions of balance and the unbalanced force from the inertia diagram. The third part of the work, by the application of a method too little used in designing, enables the exact calculations for the dimension of a part to be made, and for a check the limits between which the result should lie are presented. These limits also permit of very quick approximate design when used as empirical formulæ. Variations of the limits for size, speed, or other conditions are pointed out.

The large amount of computation and numerical work involved makes it unlikely that the book is free from errors, and I will gratefully acknowledge notice of any errors or discrepancies. In this connection I am very much indebted to my associate Prof. Amasa Trowbridge for his valuable suggestions in the reading of the manuscript.

C. E. L.

COLUMBIA UNIVERSITY, New York, Sept. 1904.

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GAS-ENGINE DESIGN.

PART I.

POWER—EFFICIENCY—ECONOMY.

1. Introduction.—It is from a comparatively simple though comprehensive statement of a guarantee to a buyer that the engineer must design and construct his engine. This statement of guarantee is the initial specification for which the designer must elaborate the details with sufficient minuteness to enable workmen to construct the machine.

Five facts are of prime importance to the buyer: first, the capacity of the engine in brake horse-power; second, its performance in B.T.U. per hour necessary to maintain each B.H.P.; third, the adaptability of the engine to the service. This is a factor of no little importance to the buyer, but is not always a feature of guarantees. When it is not, it certainly has weight in the decision of the buyer with what builder he will place his order. It may be that a certain firm has a reputation for building a better balanced engine than another, either because of owning certain useful patents or because of employing better designers; another firm may be in a position to build an engine with better speed regulation. To the former the boat-builder or owner whose building must not be shaken will apply, to the latter the buyer who intends to run dynamos in parallel. When incorporated in the guarantee this adaptability clause is mainly confined to regulation of engines to run dynamos.

The last two factors that concern the designer in the closing of the contract are the cost and workmanship. Cost is the last thing also that is learned by either designer or builder, and is determined only by experience in manufacture, not by any abstract or deep knowledge of technology, while workmanship is likewise difficult to predict.

The building of an engine to meet requirements of power economy or adaptability in balance or regulation is a thing that can be set down as a technical problem and solutions found from endless comparisons of what by mathematical analysis should be under ideal conditions and the determination by test of what is or can be under practical conditions. It is the business of the designer to know all the available results of such comparisons, for these enable him to take cognizance of conditions that are unknown or unknowable and thus to design for any specifications that are within possibility and to know just as surely what cannot be done.

Power depends on mean effective pressure and piston speed. Mean effective pressure depends in gas-engines primarily on the kind of gas and secondly on the treatment of the gas. Piston speeds in gas-engines depend on nearly the same conditions as in steam-engines.

Economy is a factor in gas-engines that depends on the kind of fuel and its treatment.

Regulation depends on the method of governing, the combination of cylinders or treatment of the gas, to produce uniformity of turning effort and permit of quick change to meet loads.

Balance in gas-engines is controlled by the same laws as in other machines. The dimensions of the parts which are covered in the guarantee under the head of workmanship depend on a knowledge of the stresses in all the parts and the resisting power of the various metals.

The stresses in gas-engines depend primarily on the kind of gas and the treatment, and secondly on the inertia of parts, just as in other engines.

It thus appears that the design of a gas engine to meet a given guarantee involves many problems of steam engine or ordinary machine design, but a knowledge of these alone is useless; it must be supplemented with the data on the behavior of gas when mixed with air and burned by the explosion method in a water-jacketed cylinder.

It is intended, after the first brief treatment of the condition surrounding the output or power and the economy, to consider the forces produced in the gas-engine cylinder, their variation throughout the stroke, and the effects of these forces on the other parts of the mechanism. The forces due to mass and motion will be next treated for balance, their combination with the forces due to the gas pressure behind the piston, for turning effort and strength in parts.

The analysis of stresses in the separate parts and their dimensions as determined by present-day practice, furnish the main part of the work and are based naturally on what precedes.

2. **Power.**—Capacity of gas-engines is generally guaranteed in terms of brake horse-power to be determined by test. This depends on indicated horse-power and the energy lost in the engine by the friction of one metal surface on another or through various gas losses in leakage, or the passage of gas through restricted openings in suction, exhaust, transfer passages, or otherwise. These losses are seldom remembered as such, but rather as involved in mechanical efficiency as defined below.

Let B.H.P. = brake horse-power;
I.H.P. = indicated horse-power;
E. = mechanical efficiency.

Then

$$E=\frac{\text{B.H.P.}}{\text{I.H.P.}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (1)$$

or

$$\text{B.H.P.} = E \times \text{I.H.P.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (2)$$

3. **Engine Losses.**—In gas-engines this mechanical efficiency depends on the extent of the different losses in the engine itself, the most important of which are, for four-cycle engines:

1° On mechanical friction, and
2° On fluid losses and leakage.

Let L_m = the fraction of indicated horse-power due to mechanical friction losses;
L_f = the fraction of indicated horse-power due to fluid losses and leakage.

Then

$$E = 1 - L_m - L_f, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (3)$$

and

$$\text{B.H.P.} = (1 - L_m - L_f)\ \text{I.H.P.} \quad . \quad . \quad . \quad . \quad . \quad . \qquad (4)$$

The mechanical friction loss in most engines is practically constant with load, but depending chiefly for its amount on lubrication and the presence or absence of peculiar energy-absorbing parts with great friction coefficients and loads. The loss due to fluid friction is in most engines quite small, about 6% of the I.H.P. when large free openings are provided, leakage is reasonably small, and the velocity of gases through openings for admission, exhaust, or transfer of gases low. It may become very high, as much as 30% in a leaky or highly throttled engine Very often mechanical efficiency is given as the simple ratio of formula (1), and as such is variable with load and speed. This is especially true on throttling engines, and when so given is apt to be somewhat misleading, since it is principally the term L_f which varies, increasing in fact very rapidly with decrease of load.

Two-cycle engines involve a much greater fluid loss than four-cycle engines in general, because of the necessity for quick transfer of the charge from the suction-chamber or precompression-pump to the motor cylinder, while the crank-pin passes through a small arc about the dead-center. To bring about this transfer of mixture and displacement of burnt gases requires a precompression of the full charge to a pressure of from 5 to 10 pounds per square inch. This entire work of precompression is lost so far as producing work is concerned, as its use is simply to introduce the charge, and the loss is the price paid for the method of working.

Call this precompression loss ($L_c \times$I.H.P.) horse-power and there results for mechanical efficiency E' of a two-cycle engine

$$E' = 1 - L_m - L_f - L_c. \quad \ldots\ldots \quad (5)$$

The resulting output will then be

$$\text{B.H.P.} = (1 - L_m - L_f - L_c)\ \text{I.H.P.} \quad \ldots\ldots \quad (6)$$

This quantity L_c will have values varying from .10 to .30, the latter for larger engines with separate pumps for air and gas, while the lower value is for simplest and most efficient pumps with lowest precompression pressures.

Besides the losses noted in the engine, there are other deductions to be made for special cases. Some forms of scavenging engines require power for their air-blasts, some engines require power for driving pump for circulating water or dynamo-driving to generate ignition current. All these special losses are not to be here considered. The efficiencies given below are for fair average cases; values higher and lower are frequently found for particularly good or particularly bad combinations.

	Mechanical Efficiency.	
Engine.	Four-cycle.	Two-cycle.
Large, 500 I.H.P. and over	.81 to .86	.63 to .70
Medium, 25 to 500 I.H.P.	.79 to .81	.64 to .66
Small, 4 to 25 I.H.P.	.74 to .80	.63 to .70

4. **Fluid Friction Losses.**—An example of how the throttling of an engine by its governor may apparently decrease its mechanical efficiency is found in the figures of Prof. Robertson on a 550-H.P. Westinghouse engine, who reports a drop from $E=85\%$ at full load to $E=63\%$ at light load. A curve of E with load for a similar 25″×30″ three-cylinder throttle governed engine is given in Fig. 1.

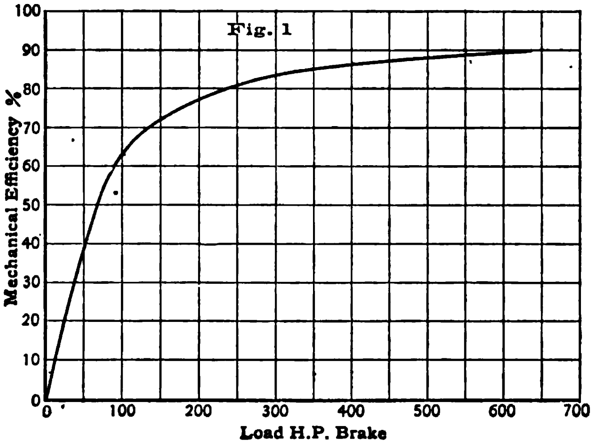


Fig. 1

If the engine friction were constant and all fluid friction also constant mechanical efficiency would be constant and the curve of Fig. 1 would become that of Fig. 2, line A. As a matter of fact the mechanical friction does decrease slightly with load. Supposing the curve B, Fig. 2, to represent this, then the difference between the ordinates of curve B and the observed curve of $\frac{\text{B.H.P.}}{\text{I.H.P.}}$

given in curve *C* will represent the fluid friction effect, which naturally is very much greater at light loads than at full load with a throttled inlet control. Hit-or-miss engines and variable cut-off governed engines suffer a very much less fluid loss than do throttled governed engines and the loss varies much less with speed, because the valve openings are constant. Leakage losses can be measured on miss strokes by finding the area between the compression and expansion lines, which of course should coincide if the walls are cold. Admission losses are measurable approximately by the area of the bottom loop on a low spring card, or, as it is often called, a pumping diagram. This is the shaded area of Fig. 3. Back-pressure fluid resistance due to too small a valve opening is also approximately measurable in the same way by determining the shaded area of Fig. 4, for example, which shows a card from a throttled engine with considerable back pressure. Incidentally in this diagram there is a loss of charge measurable by noting where the compression line crosses atmosphere at *A*. If the temperature does not change

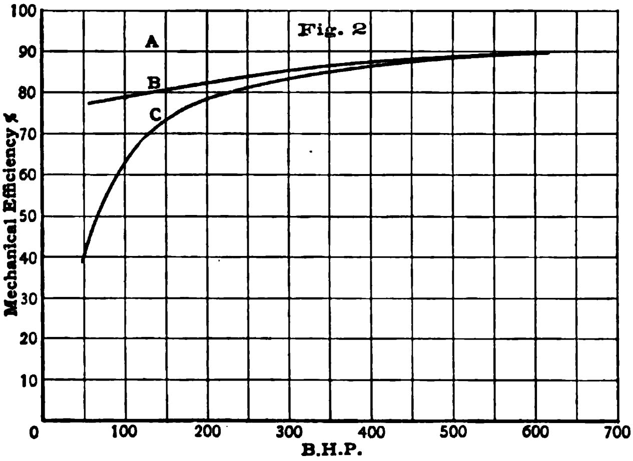


Fig. 2

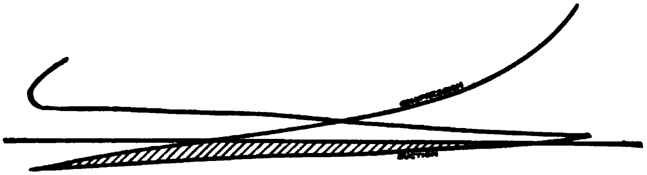


Fig. 3.

the lost charge is $\frac{BA}{BC}$, which in this case is nearly 44%. As conditions in different engines vary so greatly in these matters which are controllable by the operator as well as the designer, at least to some extent, no data can be given that would not be dangerous in their possible misapplication. However, the nature of the losses has been pointed out, and one set of curves on a large engine will be given to illustrate another point: the variation of bottom loop loss with speed. It must be remembered though that the results are applicable only to a precisely similar engine.

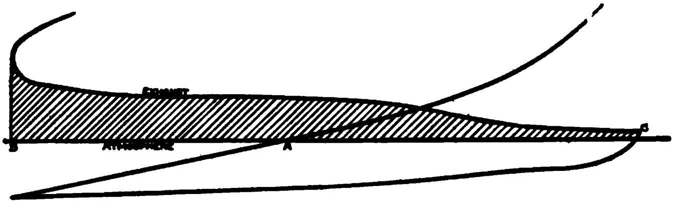


Fig. 4.

A pair of curves, Fig. 5, given by Mr. H. A. Humphrey shows the fluid friction loss, $L_f \times$I.H.P., in a 400-H.P. Crossley engine when running light and when exploding every charge under the governor. They are obtained from measuring the lost area on the bottom loops of the indicator cards showing suction and exhaust lines and part of compression, the rest being cut out by the stop

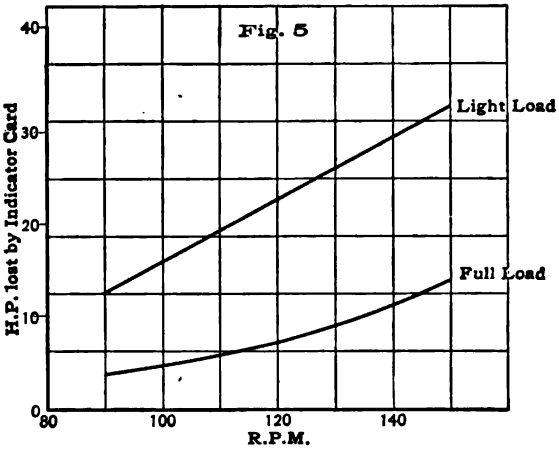


Fig. 5

placed in the indicator to save it when using the weak 32-lb. spring. These curves do not represent all of the loss L_f, but only part of it, and show that it varies with the speed.

5. Displacement for Required I.H.P.—From what precedes it is possible to determine what indicated horse-power will probably be required for the brake horse-power specified; the next step is to provide sufficient cylinder to give this I.H.P. with whatever mean effective pressures will be available with the fuel to be used.

Let P = mean effective pressure in pounds per square inch for the working stroke;
L = length of stroke in feet;
a = effective area of piston in square inches;
N = revolutions per minute;
n = explosions or impulses per minute;
S = piston speed in feet per minute, assuming piston velocity constant at all points of stroke.

Then

$$\text{I.H.P.} = \frac{PLan}{33000}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

For a four-cycle single-acting cylinder............ $n = \frac{N}{2}$,

For a two-cycle single-acting cylinder............ $n = N$,

and the horse-power will be that for one cylinder exploding one end only. For combinations of cylinders, appropriate factors must be used, remembering that for double-acting engines (a) is area of piston less that of rod.

6. Piston Speeds.—The length of stroke in feet L is covered twice in one revolution; hence $2L$ feet is covered in one revolution, or $2LN$ feet per minute, or

$$2LN = S,$$

or

$$LN = \frac{S}{2};$$

whence from formula (7), by substitution, for a single-acting four-cycle cylinder,

$$\text{I.H.P.} = \frac{1}{4}\,\frac{PaS}{33000}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Piston speeds in gas-engines vary not much from steam practice. The following may be taken as giving fair values representing the limits of practice:

I.H.P.	Piston Speed = S feet per minute.	
1000 stationary	(700–1000)	800 average
700 ‘‘	(700– 900)	750 ‘‘
500 ‘‘	(650– 850)	700 ‘‘
150 ‘‘	(600– 800)	650 ‘‘
50 ‘‘	(500– 700)	600 ‘‘
Small ‘‘	(450– 700)	550 ‘‘
2–10 H.P. (per cylinder) automobile	(600–1000)	750 ‘‘

The relation between piston speed and R.P.M. for different strokes is given in Table I, which will be found useful in saving multiplication. The credit for calculating this and other H.P. tables to follow is due to "Power," where they appeared in 1901.

7. I.H.P per Pound M.E.P.—The three quantities, piston speed, mean effective pressure, and I.H.P., have a certain relation for every cylinder diameter. These relations can be expressed in tables by assuming any one unity; thus a useful relation is indicated horse-power per pound P for all cylinder diameters and piston speed, and for a single-acting four-cycle cylinder it is given by

$$\text{I.H.P. per pound M.E.P.} = \frac{1}{4}\frac{a \times S}{33000}, \quad . \quad . \quad . \quad . \qquad (9)$$

which becomes

$$\text{I.H.P. per pound M.E.P.} = \frac{1}{2}\frac{a \times S}{33000}, \quad . \quad . \quad . \qquad (10)$$

for a single-acting two-cycle cylinder.

Table II gives the part $\frac{a \times S}{33000}$ for all diameters and piston speed.

The use of the table may be illustrated by an example:

What horse-power would be developed by a double-acting two-cycle engine 24″×48″ with a 4⅜″ piston-rod running at 620 feet of piston speed and with 46.5 pounds of mean effective pressure in the head end and 47.2 in the crank end?

TABLE I.

PISTON SPEED FEET PER MINUTE.

$$S=\frac{2\times\text{stroke in inches}\times\text{R.P.M.}}{12}$$

Stroke in Inches.	R.P.M.								
	10	20	30	40	50	60	70	80	90
1	1.67	3.33	5	6.67	8.33	10	11.67	13.33	15
2	3.33	6.67	10	13.33	16.67	20	23.33	26.67	30
3	5	10	15	20	25	30	35	40	45
4	6.67	13.33	20	26.67	41.67	40	46.67	53.33	60
5	8.33	16.67	25	33.33	50.00	50	58.33	66.67	75
6	10	20	30	40	58.33	60	70	80	90
7	11.67	23.33	35	46.67	66.67	70	81.67	93.33	105
8	13.33	26.67	40	53.33	75.00	80	93.33	106.67	120
9	15	30	45	60	83.33	90	105	120	135
10	16.67	33.33	50	66.67	91.67	100	116.67	133.33	150
11	18.33	36.67	55	73.33	100.00	110	128.33	146.67	165
12	20	40	60	80	116.67	120	140	160	180
13	21.67	43.33	65	86.67	125.00	130	151.67	173.33	195
14	23.33	46.67	70	93.33	133.33	140	163.33	186.67	210
15	25	50	75	100	141.67	150	175	200	225
16	26.67	53.33	80	106.67	150.00	160	186.67	213.33	240
17	28.33	56.67	85	113.33	158.33	170	198.33	226.67	255
18	30	60	90	120	166.67	180	210	240	270
19	31.67	63.33	95	126.67	183.33	190	221.67	253.33	285
20	33.33	66.67	100	133.33	200.00	200	233.67	266.67	300
22	36.67	73.33	110	146.67	216.67	220	256.67	293.33	330
24	40	80	120	160	233.33	240	280	320	360
26	43.33	86.67	130	173.33	250.00	260	303.33	346.67	390
28	46.67	93.33	140	186.67	233.33	280	326.67	373.33	420
30	50	100	150	200	250	300	350	400	450
32	53.33	106.67	160	213.33	266.67	320	373.33	426.67	480
34	56.67	113.33	170	226.67	283.33	340	396.67	453.33	510
36	60	120	180	240	300	360	420	480	540
38	63.33	126.67	190	253.33	316.67	380	443.33	506.67	570
40	66.67	133.33	200	266.67	333.33	400	466.67	533.33	600
42	70	140	210	280	350	420	490	560	630
44	73.33	146.67	220	293.33	366.67	440	513.33	586.67	660
46	76.67	153.33	230	306.67	383.33	460	536.67	613.33	690
48	80	160	240	320	400	480	560	640	720
50	83.33	166.67	250	333.33	416.67	500	583.33	666.67	750
52	86.67	173.33	260	346.67	433.33	520	606.67	693.33	780
54	90	180	270	360	450	540	630	720	810
56	93.33	186.67	280	373.33	466.67	560	653.33	746.67	840
58	96.67	193.33	290	386.67	483.33	580	676.67	773.33	870
60	100	200	300	400	500	600	700	800	900

Table I—(*Continued*).

PISTON SPEED, FEET PER MINUTE.

Stroke in Inches.	R.P.M.								
	10	20	30	40	50	60	70	80	90
62	103.33	206.67	310	413.33	516.67	620	723.33	826.67	930
64	106.67	213.33	320	426.67	533.33	640	746.67	853.33	960
66	110	220	330	440	550	660	770	880	990
68	113.33	226.67	340	453.33	566.67	680	793.33	906.67	1020
70	116.67	233.33	350	466.67	583.37	700	816.67	933.33	1050
72	120	240	360	480	600	720	840	960	1080
74	123.33	246.67	370	493.33	616.67	740	863.33	986.67	1110
76	126.67	253.33	380	506.67	633.33	760	886.67	1013.33	1140
78	130	260	390	520	650	780	910	1040	1170
80	133.33	266.67	400	533.33	666.67	800	933.33	1066.67	1200
82	136.67	273.33	410	546.67	683.33	820	956.67	1093.33	1230
84	140	280	420	560	700	840	980	1120	1260
86	143.33	286.67	430	573.33	716.67	860	1003.33	1146.67	1290
88	146.67	293.33	440	586.67	733.33	880	1026.67	1173.33	1320
90	150	300	450	600	750	900	1050	1200	1350
92	153.33	306.67	460	613.33	766.67	920	1073.33	1226.67	1380
94	156.67	313.33	470	626.67	783.33	940	1073.33	1253.33	1410
96	160	320	480	640	800	960	1120	1280	1440
98	163.33	326.67	490	653.33	816.67	980	1143.33	1306.67	1470
100	166.67	333.33	500	666.67	883.33	1000	1166.67	1333.33	1500
102	170	340	510	680	850	1020	1190	1360	1530
104	173.33	346.67	520	693.33	866.67	1040	1213.33	1386.67	1560
106	176.33	353.33	530	706.67	883.33	1060	1236.67	1413.33	1590
108	180	360	540	720	900	1080	1260	1440	1620
110	183.33	366.67	550	733.33	916.67	1100	1283.33	1466.67	1650
112	186.67	373.33	560	746.67	933.33	1120	1306.67	1493.33	1680
114	190	380	570	760	950	1140	1330	1520	1710
116	193.33	386.67	580	773.33	966.67	1160	1353.33	1546.67	1740
118	196.67	393.33	590	786.67	983.33	1180	1376.67	1573.33	1770
120	200	400	600	800	1000	1200	1400	1600	1800
122	203.33	406.67	610	813.33	1016.67	1220	1423.33	1626.67	1830
124	206.67	413.33	620	826.67	1033.33	1240	1446.67	1653.33	1860
126	210	420	630	840	1050	1260	1470	1680	1890
128	213.33	426.67	640	853.33	1066.67	1280	1493.33	1706.67	1920
130	216.67	433.33	650	886.33	1083.33	1300	1516.67	1733.33	1950
132	220	440	660	880	1100	1320	1540	1760	1980
134	223.33	446.67	670	893.33	1116.67	1340	1563.33	1786.67	2010
136	226.67	453.33	680	906.67	1133.33	1360	1586.67	1813.33	2040
138	230	460	690	920	1150	1380	1610	1840	2070
140	233.33	466.67	700	933.33	1166.67	1400	1633.33	1866.67	2100
142	236.67	473.33	710	946.67	1183.33	1420	1656.67	1893.33	2130
144	240	480	720	960	1200	1440	1680	1920	2160
146	243.33	486.67	730	973.33	1216.67	1460	1703.33	1946.67	2190
148	246.67	493.33	740	986.67	1233.33	1480	1726.67	1973.33	2220
150	250	500	750	1000	1250	1500	1750	2000	2250

TABLE II.

Values of $\frac{a \times S}{33000}$.

Diameter of Cylinder or Rod, Inches.	Piston Speed in Feet per Minute.								
	100	200	300	400	500	600	700	800	9000
3/4	.0013	.0027	.0040	.0054	.0067	.0080	.0094	.0107	.0120
13/16	.0016	.0031	.0047	.0063	.0079	.0094	.0110	.0126	.0141
7/8	.0018	.0036	.0055	.0073	.0091	.0109	.0128	.0146	.0164
15/16	.0021	.0042	.0063	.0084	.0105	.0126	.0146	.0167	.0188
1	.0024	.0048	.0071	.0095	.0119	.0143	.0167	.0190	.0214
1 1/16	.0027	.0054	.0081	.0107	.0134	.0161	.0188	.0215	.0242
1 1/8	.0030	.0060	.0090	.0120	.0151	.0181	.0211	.0241	.0271
1 3/16	.0034	.0067	.0101	.0134	.0168	.0201	.0235	.0268	.0302
1 1/4	.0037	.0074	.0112	.0149	.0186	.0223	.0260	.0298	.0335
1 5/16	.0041	.0082	.0123	.0164	.0205	.0246	.0287	.0328	.0369
1 3/8	.0045	.0090	.0135	.0180	.0225	.0270	.0315	.0360	.0405
1 7/16	.0049	.0098	.0148	.0197	.0246	.0295	.0344	.0393	.0443
1 1/2	.0054	.0107	.0161	.0214	.0268	.0321	.0375	.0428	.0482
1 9/16	.0058	.0116	.0174	.0232	.0291	.0340	.0407	.0465	.0523
1 5/8	.0063	.0126	.0189	.0251	.0314	.0377	.0440	.0503	.0566
1 11/16	.0068	.0136	.0203	.0271	.0339	.0407	.0474	.0542	.0610
1 3/4	.0073	.0146	.0219	.0292	.0364	.0437	.0510	.0583	.0656
1 13/16	.0078	.0156	.0235	.0313	.0391	.0469	.0548	.0626	.0704
1 7/8	.0084	.0167	.0251	.0335	.0418	.0502	.0586	.0669	.0753
1 15/16	.0089	.0179	.0268	.0357	.0447	.0536	.0625	.0715	.0804
2	.0095	.0190	.0286	.0381	.0476	.0571	.0666	.0762	.0857
2 1/16	.0101	.0202	.0304	.0405	.0506	.0607	.0709	.0810	.0911
2 1/8	.0107	.0215	.0322	.0430	.0537	.0645	.0752	.0860	.0967
2 3/16	.0114	.0228	.0342	.0456	.0569	.0683	.0797	.0911	.1025
2 1/4	.0120	.0241	.0361	.0482	.0602	.0723	.0843	.0964	.1084
2 5/16	.0127	.0255	.0382	.0509	.0636	.0764	.0891	.1018	.1145
2 3/8	.0134	.0268	.0403	.0537	.0671	.0805	.0940	.1074	.1208
2 7/16	.0141	.0283	.0424	.0566	.0707	.0848	0990	.1131	.1273
2 1/2	.0149	.0297	.0446	.0595	.0744	.0892	.1041	.1190	.1339
2 9/16	.0156	.0313	.0469	.0625	.0781	.0938	.1094	.1250	.1407
2 5/8	.0164	.0328	.0492	.0656	.0820	.0984	.1148	.1312	.1476
2 11/16	.0172	.0344	.0516	.0688	.0860	.1031	.1203	.1375	.1547
2 3/4	.0180	.0360	.0540	.0720	.0900	.1080	.1260	.1440	.1620
2 13/16	.0188	.0377	.0565	.0753	0941	.1130	.1318	.1506	.1694
2 7/8	.0197	.0393	.0590	.0787	.0984	.1180	.1377	.1574	.1770
2 15/16	.0205	.0411	.0616	.0821	.1027	.1232	.1438	.1643	.1848
3	.0214	.0428	.0643	.0857	.1071	.1285	.1499	.1714	.1928
3 1/8	.0232	.0465	.0697	.0930	.1162	.1395	.1627	.1859	.2092
3 1/4	.0251	.0503	.0754	.1006	.1257	.1508	.1760	.2011	.2262
3 3/8	.0271	.0542	.0813	.1084	.1355	.1627	.1898	.2169	.2440
3 1/2	.0292	.0583	.0875	.1166	.1458	.1740	.2041	.2332	.2624
3 5/8	.0313	.0626	.0938	.1251	.1564	.1877	.2189	.2502	.2815
3 3/4	.0335	.0669	.1004	.1339	.1673	.2008	.2343	.2678	.3012
3 7/8	.0357	.0715	.1072	.1429	.1787	.2144	.2502	.2859	.3216
4	.0381	.0762	.1142	.1523	.1904	.2285	.2066	.3046	.3427
4 1/8	.0405	.0810	.1215	.1620	.2025	.2430	.2835	.3240	.3645
4 1/4	.0430	.0860	.1290	.1720	.2149	.2579	.3009	.3439	.3869
4 3/8	.0456	.0911	.1367	.1822	.2278	.2733	.3189	.3644	.4100
4 1/2	.0482	.0964	.1446	.1928	.2410	.2892	.3374	.3856	.4337
4 5/8	.0509	.1018	.1527	.2036	.2545	.3055	.3564	.4073	.4582
4 3/4	.0537	.1074	.1612	.2149	.2686	.3223	.3760	.4298	.4835
4 7/8	.0566	.1131	.1697	.2262	.2828	.3394	.3950	.4525	.5090

TABLE II—(*Continued*).

Values of $\frac{a \times S}{33000}$.

Diameter of Cylinder or Rod, Inches.	Piston Speed in Feet per Minute.								
	100	200	300	400	500	600	700	800	900
5	.0595	.1190	.1785	.2380	.2975	.3570	.4165	.4760	.5355
5¼	.0656	.1312	.1968	.2624	.3280	.3936	.4592	.5248	.5904
5½	.0720	.1440	.2160	.2880	.3600	.4320	.5040	.5760	.6479
5¾	.0787	.1574	.2361	.3148	.3934	.4721	.5508	.6295	.7082
6	.0857	.1714	.2570	.3427	.4284	.5141	.5998	.6854	.7711
6¼	.0930	.1859	.2789	.3719	.4648	.5578	.6508	.7438	.8367
6½	.1006	.2011	.3017	.4022	.5028	.6033	.7039	.8044	.9050
6¾	.1084	.2169	.3253	.4338	.5422	.6506	.7591	.8675	.9760
7	.1166	.2332	.3499	.4665	.5831	.6997	.8163	.9330	1.0496
7¼	.1251	.2502	.3753	.5004	.6255	.7506	.8757	1.0008	1.1259
7½	.1339	.2678	.4016	.5355	.6694	.8033	.9371	1.0710	1.2049
7¾	.1429	.2859	.4288	.5718	.7147	.8577	1.0006	1.1436	1.2865
8	.1523	.3046	.4570	.6093	.7616	.9139	1.0662	1.2185	1.3709
8¼	.1620	.3240	.4860	.6480	.8099	.9719	1.1339	1.2959	1.4579
8½	.1720	.3439	.5159	.6878	.8598	1.0317	1.2037	1.3756	1.5476
8¾	.1822	.3644	.5467	.7289	.9111	1.0933	1.2755	1.4577	1.6400
9	.1928	.3856	.5783	.7711	.9639	1.1567	1.3495	1.5422	1.7350
9¼	.2036	.4073	.6109	.8146	1.0182	1.2218	1.4255	1.6291	1.8328
9½	.2148	.4296	.6444	.8592	1.0740	1.2888	1.5036	1.7184	1.9331
9¾	.2262	.4525	.6787	.9050	1.1312	1.3575	1.5837	1.8100	2.0362
10	.2380	.4760	.7140	.9520	1.1900	1.4280	1.6660	1.9040	2.1420
10½	.2624	.5248	.7872	1.0496	1.3120	1.5744	1.8368	2.0992	2.3615
11	.2880	.5760	.8639	1.1519	1.4399	1.7279	2.0159	2.3038	2.5918
11½	.3148	.6295	.9443	1.2590	1.5738	1.8885	2.2033	2.5181	2.8328
12	.3427	.6855	1.0282	1.3709	1.7136	2.0564	2.3991	2.7418	3.0845
13	.4022	.8044	1.2066	1.6088	2.0111	2.4133	2.8155	3.2177	3.6199
14	.4665	.9330	1.3995	1.8659	2.3324	2.7989	3.2654	3.7319	4.1984
15	.5355	1.0710	1.6065	2.1419	2.6774	3.2129	3.7484	4.2839	4.8194
16	.6093	1.2185	1.8278	2.4371	3.0464	3.6556	4.2649	4.8742	5.4835
17	.6878	1.3756	2.0635	2.7513	3.4391	4.1269	4.8147	5.5025	6.1904
18	.7711	1.5422	2.3134	3.0845	3.8556	4.6267	5.3978	6.1690	6.0401
19	.8592	1.7184	2.5775	3.4367	4.2959	5.1551	6.0143	6.8735	7.7326
20	.9520	1.9040	2.8560	3.8080	4.7600	5.7120	6.6640	7.6160	8.5680
21	1.0496	2.0992	3.1487	4.1983	5.2479	6.2975	7.3470	8.3966	9.4462
22	1.1519	2.3038	3.4557	4.6076	5.7595	6.9115	8.0634	9.2153	10.3672
23	1.2590	2.5181	3.7771	5.0361	6.2952	7.5542	8.8132	10.0722	11.3313
24	1.3709	2.7418	4.1126	5.4835	6.8544	8.2253	9.5962	10.9670	12.3379
25	1.4875	2.9750	4.4625	5.9499	7.4374	8.9249	10.4124	11.8999	13.3874
26	1.6089	3.2178	4.8266	6.4355	8.0444	9.6533	11.2622	12.8710	14.4799
27	1.7350	3.4701	5.2051	6.9401	8.6752	10.4102	12.1452	13.8802	15.6153
28	1.8659	3.7318	5.5977	7.4636	9.3295	11.1955	13.0614	14.9273	16.7932
29	2.0016	4.0032	6.0047	8.0063	10.0079	12.0095	14.0110	16.0126	18.0142
30	2.1420	4.2840	6.4260	8.5680	10.7100	12.8520	14.9940	17.1360	19.2780
31	2.2872	4.5744	6.8615	9.1487	11.4359	13.7231	16.0103	18.2975	20.5846
32	2.4371	4.8742	7.3114	9.7485	12.1856	14.6227	17.0598	19.4970	21.9341
33	2.5918	5.1836	7.7755	10.3673	12.9591	15.5509	18.1427	20.7345	23.3264
34	2.7513	5.5025	8.2538	11.0051	13.7564	16.5076	19.2589	22.0102	24.7615
35	2.9155	5.8310	8.7465	11.6619	14.5774	17.4929	20.4084	23.3239	26.2394
36	3.0845	6.1690	9.2535	12.3379	15.4224	18.5069	21.5914	24.6759	27.7604
37	3.2582	6.5164	9.7746	13.0328	16.2911	19.5493	22.8075	26.0657	29.3239
38	3.4367	6.8734	10.3101	13.7468	17.1835	20.6202	24.0569	27.4936	30.9303
39	3.6200	7.2399	10.8599	14.4799	18.0998	21.7198	25.3398	28.9598	32.5797
40	3.8080	7.6160	11.4240	15.2320	19.0400	22.8480	26.6560	30.4640	34.2720
41	4.0008	8.0015	12.0023	16.0030	20.0038	24.0045	28.0053	32.0061	36.0068
42	4.1983	8.3966	12.5949	16.7932	20.9015	25.1898	29.3881	33.5864	37.7847
43	4.4006	8.8012	13.2018	17.6024	22.0030	26.4036	30.8042	35.2048	39.6055
44	4.6077	9.2153	13.8230	18.4307	23.0383	27.6460	32.2537	36.8613	41.4690

TABLE II—(*Continued*).

Values of $\frac{a \times S}{33000}$.

Diameter of Cylinder or Rod, Inches.	Piston Speed in Feet Per Minute.								
	100	200	300	400	500	600	700	800	900
45	4.8195	9.6390	14.4585	19.2779	24.0974	28.9161	33.7364	38.5559	43.3754
46	5.0361	10.0721	15.1082	20.1442	25.1803	30.2164	35.2524	40.2885	45.3245
47	5.2574	10.5148	15.7722	21.0296	26.2870	31.5444	36.8018	42.0592	47.3165
48	5.4835	10.9670	16.4505	21.9341	27.4176	32.9011	38.3846	43.8681	49.3516
49	5.7144	11.4287	17.1431	22.8575	28.5718	34.2862	40.0005	45.7149	51.4293
50	5.9500	11.9000	17.8500	23.8000	29.7500	35.7000	41.6500	47.6000	53.5500
51	6.1904	12.3807	18.5711	24.7615	30.9518	37.1422	43.3325	49.5229	55.7133
52	6.4355	12.8710	19.3065	25.7421	32.1776	38.6131	45.0486	51.4841	57.9196
53	6.6854	13.3708	20.0562	26.7416	33.4270	40.1124	46.7978	53.4832	60.1685
54	6.9401	13.8801	20.8202	27.7602	34.7003	41.6404	48.5804	55.5205	62.4605
55	7.1995	14.3990	21.5985	28.7979	35.9974	43.1967	50.3964	57.5959	64.7954
56	7.4637	14.9273	22.3910	29.8547	37.3183	44.7820	52.2457	59.7093	67.1730
57	7.7326	15.4652	23.1978	30.9304	38.6630	46.3956	54.1282	61.8608	69.5935
58	8.0063	16.0126	24.0189	32.0252	40.0315	48.0378	56.0441	64.0504	72.0567
59	8.2488	16.5695	24.8543	33.1390	41.4238	49.7085	57.9933	66.2781	74.5628
60	8.5680	17.1359	25.7039	34.2719	42.8398	51.4078	59.9758	68.5438	77.1117
61	8.8560	17.7119	26.5679	35.4239	44.2798	53.1358	61.9918	70.8478	79.7037
62	9.1487	18.2974	27.4461	36.5948	45.7435	54.8922	64.0409	73.1896	82.3383
63	9.4462	18.8924	28.3386	37.7848	47.2311	56.6775	66.1235	75.5697	85.0159
64	9.7485	19.4969	29.2454	38.9938	48.7423	58.4907	68.2392	77.9876	87.7361
65	10.0555	20.1110	30.1665	40.2219	50.2774	60.3329	70.3884	80.4439	90.4994
66	10.3672	20.7345	31.1017	41.4690	51.8362	62.2035	72.5707	82.9379	93.3052
67	10.6838	21.3676	32.0514	42.7352	53.4189	64.1027	74.7865	85.4703	96.1541
68	11.0051	22.0102	33.0153	44.0204	55.0255	66.0305	77.0356	88.0407	99.0458
69	11.3312	22.6023	33.9935	45.3246	56.6558	67.9869	79.3181	90.6492	101.9804
70	11.6620	23.3239	34.9859	46.6479	58.3098	69.9718	81.6338	93.2958	104.9577
71	11.9975	23.9051	35.9926	47.9902	59.9877	71.9853	83.9828	95.9804	107.9779
72	12.3379	24.6758	37.0136	49.3515	61.6894	74.0273	86.3651	98.7030	111.0409
73	12.6830	25.3660	38.0490	50.7320	63.4150	76.0980	88.7810	101.4640	114.1470
74	13.0328	26.0057	39.0985	52.1314	65.1642	78.1971	91.2299	104.2628	117.2956
75	13.3875	26.7749	40.1624	53.5498	66.9373	80.3247	93.7122	107.0996	120.4871
76	13.7468	27.4937	41.2405	54.9874	68.7342	82.4811	96.2279	109.9748	123.7216
77	14.1110	28.2220	42.3330	56.4440	70.5550	84.6660	98.7770	112.8880	126.9990
78	14.4799	28.9598	43.4396	57.9195	72.3994	86.8793	101.3591	115.8390	130.3189
79	14.8535	29.7071	44.5606	59.4142	74.2677	89.1213	103.9748	118.8284	133.6819
80	15.2320	30.4639	45.6959	60.9279	76.1598	91.3918	106.6238	121.8558	137.0877
81	15.6152	31.2303	46.8455	62.4606	78.0758	93.6909	109.3061	124.9212	140.5364
82	16.0031	32.0062	48.0093	64.0124	80.0155	96.0185	112.0216	128.0247	144.0278
83	16.3958	32.7916	49.1874	65.5832	81.9789	98.3747	114.7705	131.1663	147.5621
84	16.7932	33.5865	50.3797	67.1730	83.9662	100.7595	117.5527	134.3459	151.1392
85	17.1955	34.3909	51.5864	68.7818	85.9773	103.1727	120.3682	137.5636	154.7591
86	17.6024	35.2048	52.8073	70.4097	88.0121	105.6145	123.2170	140.8194	158.4218
87	18.0142	36.0284	54.0425	72.0567	90.0709	108.0851	126.0993	144.1135	162.1276
88	18.4307	36.8613	55.2920	73.7227	92.1533	110.5840	129.0147	147.4453	165.8760
89	18.8519	37.7039	56.5558	75.4078	94.2597	113.1116	131.9636	150.8155	169.6675
90	19.2780	38.5559	57.8339	77.1119	96.3898	115.6678	134.9458	154.2238	173.5017
91	19.7087	39.4175	59.1262	78.8349	98.5436	118.2524	137.9611	157.6698	177.3785
72	20.1443	40.2885	60.4328	80.5771	100.7214	120.8656	141.0099	161.1542	181.2985
93	20.5846	41.1692	61.7537	82.3383	102.9229	123.5075	144.0920	164.6766	185.2612
94	21.0296	42.0593	63.0889	84.1185	105.1482	126.1778	147.2075	168.2371	189.2667
95	21.4705	42.9589	64.4384	85.9178	107.3973	128.8717	150.3562	171.8356	193.3151
96	21.9340	43.8681	65.8021	87.7361	109.6702	131.6042	153.5382	175.4722	197.4063
97	22.3934	44.7867	67.1801	89.5735	111.9668	134.3642	156.7535	179.1469	201.5403
98	22.8575	45.7149	68.5724	91.4298	114.2873	137.1447	160.0022	182.8596	205.7171
99	23.3263	46.6527	69.9790	93.3053	116.6317	139.9590	163.2843	186.6107	209.9370
100	23.7999	47.5999	71.3998	95.1998	118.9997	142.7996	166.5996	190.3995	214.1995

From the table the constant for this engine would be:

$$\begin{array}{lr} \text{Value for } 600 = & 8.2253 \\ \text{`` `` } 20 = & .27418 \\ \hline \text{`` `` } 620 = & 8.49948 \end{array}$$

horse-power per pound of average mean effective pressure.

The average mean effective pressure would be

$$\frac{46.5+47.2}{2}=46.85 \text{ pounds.}$$

The horse-power uncorrected for the rod would therefore be

$$8.49948\times 46.85=398.200638 \text{ H.P.}$$

The horse-power lost by the presence of the rod is that which would be developed by an engine 4⅜ inches in diameter at 310 feet of piston speed and at 47.2 pounds mean effective pressure. From the table we find the constant for such an engine to be:

$$\begin{array}{lr} \text{For } 300 \text{ feet,} & .1367 \\ \text{``} \quad 10 \text{ ``} & .00456 \\ \hline \text{``} \quad 310 \text{ ``} & .14126 \end{array}$$

horse-power per pound of mean effective pressure.

The mean effective pressure which would have acted upon this area is 47.2 pounds.

The horse-power to be deducted, therefore, is

$$.14126\times 47.2=6.667472 \text{ H.P.}$$

Deducting this from the uncorrected horse-power we have

$$\begin{array}{r} 398.200638 \\ 6.667472 \\ \hline 391.533166 \end{array}$$

as the horse-power corrected for the rod.

8. Displacement per I.H.P.—Another of the relations sometimes useful in making comparisons is the relation of indicated horse-power and P with cubic feet of displacement per minute.

Let A = effective area of piston in square feet; then $a = 144A$. Substituting

$$\text{I.H.P.} = \frac{1}{4} P \times \frac{144AS}{33000}.$$

But $A \times S$ = displacement in cubic feet per minute, or calling the displacement V, then for single-acting four-cycle cylinders

$$\text{I.H.P.} = \frac{1}{4} P \times \frac{144}{33000} \times V. \quad \ldots\ldots \quad (11)$$

For single-acting two-cycle cylinders

$$\text{I.H.P.} = \frac{1}{2} P \times \frac{144}{33000} \times V. \quad \ldots\ldots \quad (12)$$

From (11).

$$\text{Cubic feet displacement per I.H.P.} = 4 \times \frac{33000}{144P} = 4 \times \frac{1375}{6P} \quad . \quad (13)$$

for four-cycle, and for two-cycle the factor 4 becomes 2. The quantity $\frac{1375}{6P}$ is given in Table III.

Formula (13) gives by transposing

$$P = 4 \times \frac{1375}{6V}, \quad \ldots\ldots\ldots \quad (14)$$

which is the mean effective pressure required per I.H.P. for a four-cycle engine and making the factor 4 a 2, the relation for two-cycle engines results.

This quantity, $\frac{1375}{6V}$, is given in the last table if therein V be taken for P.

After the appropriate piston speed the next important quantity to be determined is the appropriate or possible mean effective pressure. This will vary with compression, kind of fuel, the dilution by neutral gases, the initial temperature, the proportion of parts of the mixture, the speed, the valve throttling on suction and exhaust, the governor action, points of ignition, and a host of smaller influences sometimes very hard to determine. It is then difficult to predict

TABLE III.

DISPLACEMENT PER H.P. AT DIFFERENT M.E.P.

M.E.P.	Cubic Feet per Minute.									
	0	.1	.2	.3	.4	.5	.6	.7	.8	.9
1	229.17	208.33	190.97	176.28	163.69	152.78	143.23	134.80	127.31	120.61
2	114.58	109.13	104.17	99.64	95.49	91.67	88.14	84.88	81.85	79.02
3	76.39	73.92	71.61	69.44	67.40	65.48	63.66	61.94	60.31	58.76
4	57.29	55.89	54.56	53.29	52.08	50.93	49.82	48.76	47.74	46.77
5	45.83	44.93	44.07	43.24	42.44	41.67	40.92	40.20	39.51	38.84
6	38.19	37.57	36.96	36.38	35.81	35.26	34.72	34.20	33.70	33.21
7	32.74	32.28	31.83	31.39	30.97	30.56	30.15	29.76	29.38	29.01
8	28.65	28.29	27.95	27.61	27.28	26.96	26.65	26.34	26.04	25.75
9	25.46	25.18	24.91	24.64	24.38	24.12	23.87	23.63	23.38	23.15
10	22.92	22.69	22.47	22.25	22.04	21.83	21.62	21.42	21.22	21.02
11	20.83	20.65	20.46	20.28	20.10	19.93	19.76	19.59	19.42	19.26
12	19.10	18.94	18.78	18.63	18.48	18.33	18.19	18.04	17.70	17.76
13	17.63	17.49	17.36	17.23	17.10	16.98	16.85	16.73	16.61	16.49
14	16.37	16.25	16.14	16.03	15.91	15.80	15.70	15.59	15.48	15.38
15	15.28	15.18	15.08	14.98	14.88	14.78	14.69	14.60	14.50	14.41
16	14.32	14.23	14.15	14.06	13.97	13.89	13.81	13.72	13.64	13.56
17	13.48	13.40	13.32	13.25	13.17	13.10	13.02	12.95	12.87	12.80
18	12.73	12.66	12.59	12.52	12.45	12.39	12.32	12.25	12.19	12.13
19	12.06	12.00	11.94	11.87	11.81	11.75	11.67	11.63	11.57	11.52
20	11.47	11.40	11.34	11.29	11.23	11.18	11.12	11.07	11.02	10.96
21	10.91	10.86	10.81	10.76	10.71	10.66	10.61	10.56	10.51	10.46
22	10.42	10.37	10.32	10.28	10.23	10.19	10.14	10.10	10.05	10.01
23	9.96	9.92	9.88	9.84	9.79	9.75	9.71	9.67	9.63	9.59
24	9.55	9.51	9.47	9.43	9.39	9.35	9.32	9.28	9.24	9.20
25	9.17	9.13	9.09	9.06	9.02	8.99	8.95	8.92	8.88	8.85
26	8.81	8.78	8.75	8.71	8.68	8.65	8.62	8.58	8.55	8.52
27	8.49	8.46	8.43	8.39	8.36	8.33	8.30	8.27	8.24	8.21
28	8.18	8.16	8.13	8.10	8.07	8.04	8.01	7.98	7.96	7.93
29	7.90	7.88	7.85	7.82	7.79	7.77	7.74	7.72	7.69	7.66
30	7.64	7.61	7.59	7.56	7.54	7.51	7.49	7.46	7.44	7.42
31	7.39	7.37	7.35	7.32	7.30	7.28	7.25	7.23	7.21	7.18
32	7.16	7.14	7.12	7.09	7.07	7.05	7.03	7.01	6.99	6.97
33	6.94	6.92	6.90	6.88	6.86	6.84	6.82	6.80	6.78	6.76
34	6.74	6.72	6.70	6.68	6.66	6.64	6.62	6.60	6.59	6.57
35	6.55	6.53	6.51	6.49	6.47	6.46	6.44	6.42	6.40	6.38
36	6.37	6.35	6.33	6.31	6.30	6.28	6.26	6.24	6.23	6.21
37	6.19	6.18	6.16	6.14	6.13	6.11	6.09	6.08	6.06	6.05
38	6.03	6.01	6.00	5.98	5.97	5.95	5.94	5.92	5.91	5.89
39	5.88	5.86	5.85	5.83	5.82	5.80	5.79	5.77	5.76	5.74
40	5.73	5.71	5.70	5.69	5.67	5.66	5.64	5.63	5.62	5.60
41	5.59	5.58	5.56	5.55	5.54	5.52	5.51	5.50	5.48	5.47
42	5.46	5.44	5.43	5.42	5.40	5.39	5.38	5.37	5.35	5.34
43	5.33	5.32	5.30	5.29	5.28	5.27	5.26	5.24	5.23	5.22
44	5.21	5.20	5.18	5.17	5.16	5.15	5.14	5.13	5.12	5.10
45	5.09	5.08	5.07	5.06	5.05	5.04	5.03	5.01	5.00	4.99

TABLE III—(*Continued*).
DISPLACEMENT PER H.P. AT DIFFERENT M.E.P.

M.E.P.	Cubic Feet per Minute.									
	0	.1	.2	.3	.4	.5	.6	.7	.8	.9
46	4.98	4.97	4.96	4.95	4.94	4.93	4.92	4.91	4.90	4.89
47	4.88	4.87	4.86	4.85	4.83	4.82	4.81	4.80	4.79	4.78
48	4.77	4.76	4.75	4.74	4.73	4.73	4.72	4.71	4.70	4.69
49	4.68	4.67	4.66	4.65	4.64	4.63	4.62	4.61	4.60	4.59
50	4.58	4.57	4.57	4.56	4.55	4.54	4.53	4.52	4.51	4.50
51	4.49	4.48	4.48	4.47	4.46	4.45	4.44	4.43	4.42	4.42
52	4.41	4.40	4.39	4.38	4.37	4.37	4.36	4.35	4.34	4.33
53	4.32	4.32	4.31	4.30	4.29	4.28	4.28	4.27	4.26	4.25
54	4.24	4.24	4.23	4.22	4.21	4.20	4.20	4.19	4.18	4.17
55	4.17	4.16	4.15	4.14	4.14	4.13	4.12	4.11	4.11	4.10
56	4.09	4.09	4.08	4.07	4.06	4.06	4.05	4.04	4.03	4.03
57	4.02	4.01	4.01	4.00	3.99	3.99	3.98	3.97	3.96	3.96
58	3.95	3.94	3.94	3.93	3.92	3.92	3.91	3.90	3.90	3.89
59	3.88	3.88	3.87	3.86	3.85	3.85	3.85	3.84	3.83	3.83
60	3.82	3.81	3.81	3.80	3.79	3.79	3.78	3.78	3.77	3.76
61	3.76	3.75	3.74	3.74	3.73	3.73	3.72	3.71	3.71	3.70
62	3.70	3.69	3.68	3.68	3.67	3.67	3.66	3.66	3.65	3.64
63	3.64	3.63	3.63	3.62	3.61	3.61	3.60	3.60	3.59	3.59
64	3.58	3.58	3.57	3.56	3.56	3.55	3.55	3.54	3.54	3.53
65	3.53	3.52	3.51	3.51	3.50	3.50	3.49	3.49	3.48	3.48
66	3.47	3.47	3.46	3.46	3.45	3.45	3.44	3.44	3.43	3.43
67	3.42	3.42	3.41	3.41	3.40	3.40	3.39	3.39	3.38	3.38
68	3.37	3.37	3.36	3.36	3.35	3.35	3.34	3.34	3.33	3.33
69	3.32	3.32	3.31	3.31	3.30	3.30	3.29	3.29	3.28	3.28
70	3.27	3.27	3.26	3.26	3.26	3.25	3.25	3.24	3.24	3.23
71	3.23	3.22	3.22	3.21	3.21	3.21	3.20	3.20	3.19	3.19
72	3.18	3.19	3.17	3.17	3.17	3.16	3.16	3.15	3.15	3.14
73	3.14	3.14	3.13	3.13	3.12	3.12	3.11	3.11	3.11	3.10
74	3.10	3.09	3.09	3.08	3.08	3.08	3.07	3.07	3.06	3.06
75	3.06	3.05	3.05	3.04	3.04	3.04	3.03	3.03	3.02	3.02
76	3.02	3.01	3.01	3.00	3.00	3.00	2.99	2.99	2.98	2.98
77	2.98	2.97	2.97	2.96	2.96	2.96	2.95	2.95	2.95	2.94
78	2.94	2.93	2.93	2.93	2.92	2.92	2.92	2.91	2.91	2.90
79	2.90	2.90	2.89	2.89	2.89	2.88	2.88	2.88	2.87	2.87
80	2.86	2.86	2.86	2.85	2.85	2.85	2.84	2.84	2.84	2.83
81	2.83	2.83	2.82	2.82	2.82	2.81	2.81	2.81	2.80	2.80
82	2.79	2.79	2.79	2.78	2.78	2.78	2.77	2.77	2.77	2.76
83	2.76	2.76	2.75	2.75	2.75	2.74	2.74	2.74	2.73	2.73
84	2.73	2.72	2.72	2.72	2.72	2.71	2.71	2.71	2.70	2.70
85	2.70	2.69	2.69	2.69	2.68	2.68	2.68	2.67	2.67	2.67
86	2.66	2.66	2.66	2.66	2.65	2.65	2.65	2.64	2.64	2.64
87	2.63	2.63	2.63	2.63	2.62	2.62	2.62	2.61	2.61	2.61
88	2.60	2.60	2.60	2.60	2.59	2.59	2.59	2.58	2.58	2.58
89	2.57	2.57	2.57	2.57	2.56	2.56	2.56	2.55	2.55	2.55
90	2.55	2.54	2.54	2.54	2.54	2.53	2.53	2.53	2.52	2.52

TABLE III—(*Continued*).

DISPLACEMENT PER H.P. AT DIFFERENT M.E.P.

M.E.P.	Cubic Feet per Minute.									
	0	.1	.2	.3	.4	.5	.6	.7	.8	.9
91	2.52	2.52	2.51	2.51	2.51	2.50	2.50	2.50	2.50	2.49
92	2.49	2.49	2.49	2.48	2.48	2.48	2.47	2.47	2.47	2.47
93	2.46	2.46	2.46	2.46	2.45	2.45	2.45	2.45	2.44	2.44
94	2.44	2.44	2.43	2.43	2.43	2.43	2.42	2.42	2.42	2.41
95	2.41	2.41	2.41	2.40	2.40	2.40	2.40	2.39	2.39	2.39
96	2.39	2.38	2.38	2.38	2.38	2.37	2.37	2.37	2.37	2.37
97	2.36	2.36	2.36	2.36	2.35	2.35	2.35	2.35	2.34	2.34
98	2.34	2.34	2.33	2.33	2.33	2.33	2.32	2.32	2.32	2.32
99	2.31	2.31	2.31	2.31	2.31	2.30	2.30	2.30	2.30	2.29
100	2.29	2.29	2.29	2.28	2.28	2.28	2.28	2.28	2.27	2.27

a mean effective pressure with anything like the degree of accuracy possible with steam, but the process or method in use with steam-engines will be followed here. As an offset it might be remarked that economies can be predicted better than in steam-engines.

9. **Standard Reference Diagram.**—A standard reference diagram will be established and its mean effective pressure determined. Then comparison with this for any given case will give a factor which represents the percentage of the mean effective pressure of the reference diagram realizable in the actual engine of the class involved. The factor will be called, as in steam-work, the diagram factor and may be used in future design in connection with the mean effective pressure of the reference diagram to predict the probable pressure P that will result.

The standard reference diagram will be taken as that indicator card which would result if pure air passed through the four phases of an ideal cycle, receiving the same amount of heat per pound of air working as does the pound of air and gas mixture in its combustion. Such a diagram is given in Fig. 6.

The four ideal phases are:

(*a*) Adiabatic compression during which

$$PV^{1.4}=C \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (15)$$

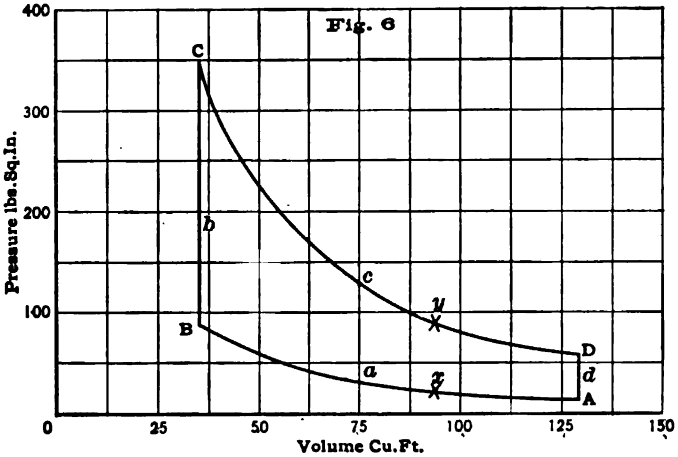
up to the same compression as is used in the engine starting at one atmosphere, 14.7 lbs. per square inch and 62° F. This assumes no loss of charge by throttling.

(*b*) Heating at constant volume $V=C$ to such a pressure as will result from addition of the heat available in one pound gas mixture to one pound air.

(*c*) Expansion adiabatically $PV^{1.4}=C$ to the same volume as before compression.

(*d*) Pressure drop to atmosphere at constant volume $V=C$ as would result either from cooling or exhausting.

Construction of reference diagram begins with point *A*, Fig. 6.



One pound air at 32° F. or 492° absolute has a volume of 12.387 cubic feet at a pressure of 14.7 lbs. per square inch; hence at 62° F. or 522° absolute

$$V_a = \frac{522}{492} \times 12.387 = 13.13 \text{ cu. ft.},$$

$$P_a = 14.7 \text{ lbs. per square inch.}$$

For any other point *x* on the compression line assume some pressure in terms of the initial pressure and find the corresponding volume by the formula

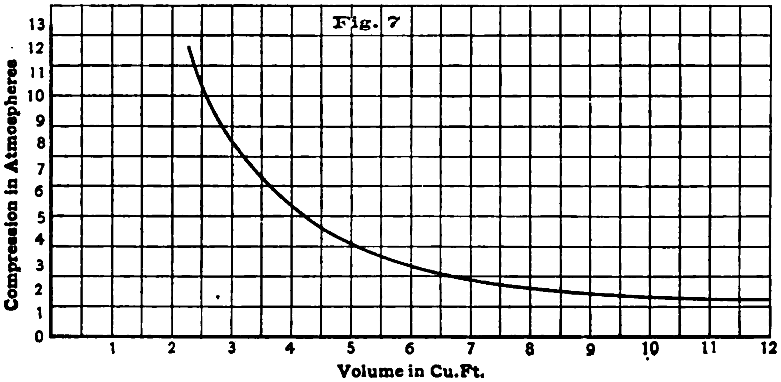
$$V_x = V_a\left(\frac{P_a}{P_x}\right)^{\frac{1}{1.4}} = 13.13\left(\frac{P_a}{P_x}\right)^{.71} \quad . \; . \; . \; . \; . \quad (16)$$

This formula can be solved by logarithms, and by this method the following table was calculated.

TABLE IV.

Compression Atmospheres, $\frac{P_x}{P_a}$.	$\left(\frac{P_a}{P_x}\right)^{.71}$.	Compression Volume, V_x. Cubic Feet.	Compression Atmospheres, $\frac{P_x}{P_a}$.	$\left(\frac{P_a}{P_x}\right)^{.71}$.	Compression Volume, V_x, Cubic Feet.
1.5	.881	11.57	7.5	.239	3.11
2.0	.611	8.02	8.0	.228	2.99
2.5	.5217	6.85	8.5	.219	2.875
3.0	.458	6.01	9.0	.209	2.74
3.5	.411	5.39	9.5	.202	2.65
4.0	.374	4.91	10.0	.195	2.56
4.5	.343	4.50	10.5	.188	2.47
5.0	.319	4.19	11.0	.182	2.39
5.5	.298	3.91	11.5	.177	2.32
6.0	.281	3.69	12.0	.171	2.245
6.5	.265	3.48	12.5	.166	2.18
7.0	.251	3.29			

These results are plotted in Fig. 7, which if cut out and used as a template facilitates the work of laying out these curves very much.



The highest compression pressure, that for *B*, should be the same as for the actual indicator card under comparison.

10. Combustion Line for Reference Diagram.—The pressure rise along *B* to *C* due to heating will be determined solely by the amount of heat given to the one pound of air.

Let Q = B.T.U. given to one pound air.

Then

$$T_c - T_b = \frac{Q}{C_v}. \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (17)$$

But

$$\frac{P_cV_c}{T_c} = \frac{P_bV_b}{T_b},$$

whence

$$\frac{P_c}{P_b}=\frac{T_c}{T_b}=\frac{Q}{C_vT_b}+1, \quad \ldots\ldots\ldots\ldots \quad (18)$$

and

$$P_c-P_b=\frac{P_b}{T_b}\frac{Q}{C_v}. \quad \ldots\ldots\ldots\ldots \quad (19)$$

But

$$\frac{P_b}{T_bC_v}=\frac{R}{V_bC_v}, \quad \text{and} \quad \left\{\begin{aligned} R &=778(C_p-C_v), \\ V_b &=V_a\left(\frac{P_a}{P_b}\right)^{\frac{1}{n}} \end{aligned}\right.$$

hence

$$\frac{P_b}{T_bC_v}=\frac{778(C_p-C_v)}{C_vV_a\left(\frac{P_a}{P_b}\right)^{\frac{1}{n}}}$$

$$=\frac{778(n-1)}{V_a\left(\frac{P_a}{P_b}\right)^{\frac{1}{n}}} \text{ where } P \text{ is in lbs. per sq. ft.}$$

$$=2.2\frac{1}{V_a\left(\frac{P_a}{P_b}\right)^{\frac{1}{n}}}. \quad \ldots\ldots\ldots\ldots \quad (20)$$

From which, by substitution in (19),

$$P_c-P_b=\frac{2.2}{\left(\frac{P_a}{P_b}\right)^{.71}}\frac{Q}{V_a} \text{ lbs. per square inch.} \quad \ldots \quad (21)$$

The ratio

$$\frac{Q}{V_a}=\frac{\text{B.T.U. per lb. mixture}}{\text{Cubic feet per lb. mixture}}$$

$$=\text{B.T.U. per cubic foot mixture.} \quad \ldots \quad (22)$$

Let H = B.T.U. per cubic ft. gas products hot;
a' = cubic ft. air to burn 1 cubic ft. gas;
n = cubic ft. neutral added;
$n+a'+1$ = cubic ft. of mixture containing 1 cubic ft. gas.

Then 1 cubic ft. mixture is made up of $\frac{1}{n+a'+1}$ cubic ft. gas and $\frac{a'}{n+a'+1}$ cu. ft. air, and the heat per cubic ft. mixture is given by

$$\frac{Q}{V_a}=\frac{H}{n+a'+1}, \quad \ldots\ldots\ldots \quad (23)$$

in which the gas shall not exceed the critical amount that the air will support.

Hence

$$P_c - P_b = \frac{2.2}{\left(\frac{P_a}{P_b}\right)^{.71}} \frac{H}{n + a' + 1}. \quad . \; . \; . \; . \; . \; . \quad (24)$$

The quantity in parenthesis can be obtained from the table already given, and the value of $\frac{H}{a' + 1}$ can be found from the table of these values, or can be calculated from the chemical composition of the fuel. If the combustion take place before compression, $P_a = P_b$; hence

$$P_c - P_a = 2.2 \frac{H}{n + a' + 1}. \quad . \; . \; . \; . \; . \; . \; . \quad (25)$$

When the neutral is to be derived from the burnt gas in the clearance, it will be best to take the volume n equal to the clearance and at same temperature as the mixture, which was 62° F.

In this case n becomes n' and

$$\frac{n'}{a' + 1} = \frac{V_b}{V_a - V_b},$$

$$n' = \frac{(V_b)\,(a' + 1)}{V_a - V_b}. \quad . \; . \; . \; . \; . \; . \; . \quad (26)$$

Hence

$$P_c - P_b = \frac{2.2}{\left(\frac{P_a}{P_b}\right)^{.71}} \frac{H}{(a' + 1)\left(1 + \frac{V_b}{V_a - V_b}\right)}$$

$$= \frac{2.2}{\left(\frac{P_a}{P_b}\right)^{.71}} \frac{H}{(a' + 1)\left(\frac{V_a}{V_a - V_b}\right)}$$

$$= 2.2 \frac{\left(1 - \left(\frac{P_a}{P_b}\right)^{.71}\right)}{\left(\frac{P_a}{P_b}\right)^{.71}} \frac{H}{a' + 1}$$

$$= 2.2 \left(\frac{1}{\left(\frac{P_a}{P_b}\right)^{.71}} - 1 \right) \frac{H}{a' + 1}. \quad . \; . \; . \; . \; . \; . \quad (27)$$

Values for $\left(\frac{P_a}{P_b}\right)^{.71}$ for any compression have been given in Table IV, while for some characteristic gases $\frac{H}{a' + 1}$ is given in Table V, but this ratio should always be calculated for each special case.

TABLE

GAS

$\frac{H}{a'+1}$ B.T.U. PER CUBIC FOOT GAS AND PER

Air required per

	H	CO	CH_4	C_2H_4
Cannel-coal gas	27.7	6.8	50.0	13.0
Common-coal gas	39.78	7.04	45.16	6.38
“ “	47.9	6.0	33.3	12.3
“ “	46.0	7.5	39.5	3.8
“ “	50.1	6.0	38.0	4.0
“ “	47.73	6.15	35.6	4.88
“ “	52.9	7.18	31.80	5.0
“ “	52.5	5.0	34.0	4.0
“ “	44.4	5.2	37.1	2.3
Carbureted water-gas	21.8	28.1	30.7	12.9
“ “	18.4	14.8	30.7	21.2
“ “	22.6	29.2	31.9	3.4
“ “	29.6	28.33	24.42	16.12
“ “	34.0	33.0	15.0	12.5
Coke-oven gas	53.0	6.0	35.0	2.0
Oil-gas *	5.6	8.9	54.9	28.9
“	24.3	.....	58.3	17.4
Natural gas	1.7	0.55	94.16	0.30
“ “	1.4	0.60	93.57	0.15
“ “	2.14	0.44	93.85	0.20
“ “	1.84	0.41	93.35	0.35
“ “	2.01	0.73	93.07	0.47
“ “	2.50	0.40	92.67	0.25
“ “	2.18	0.50	92.6	0.31
Uncarbureted water-gas	49.55	45.89	.....	.....
“ “	51.8	43.4	.....	.....
“ “	49.17	43.75	0.31	.....
“ “	49.50	35.93	1.05	.....
Producer-gas, little steam	9.2	25.3	3.1	0.8
“ “ “	9.8	24.0	3.4	0.4
“ “ “	8.37	22.74	2.56	0.36
“ “ “	6.9	20.8	2.2	0.20
“ “ “	8.5	24.8	5.2	0.40
“ “ “	6.8	22.1	3.74	0.34
“ “ “	5.3	20.0	3.0	0.20
Loomis Pettibone coal	14.0	20.0	2.0	0.20
“ “ wood	14.0	20.0	2.0	0.20
Dowson gas, average	18.0	25.0	3.0	.....
Taylor gas, average	21.0	12.0	2.0	.....
Mond gas	29.0	12.0	2.0	.....
“ “	28.5	11.5	2.1	.....
Blast-furnace gas	3.0	27.5	.....	.....

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* Gasolene vapor when not used in carburetors requiring vacuum, and kerosene vapor when not injected as liquid or requiring vacuum for aspiration, may be taken the same as this oil-gas.

V.

CHARACTERISTICS.

CUBIC FOOT OF MIXTURE, HIGH AND LOW.

cubic foot of gas.

O	CO_2	N	Cubic Feet of Air per Cubic Foot of Gas.	Gas, B.T.U. per Cubic Foot.		Mixture, B.T.U. per Cubic Foot.	
				High.	Low.	$\frac{H}{a'+1}$ High.	$\frac{H}{a'+1}$ Low.
....	0.1	2.4	6.50	843	762	98.2	85.7
0.06	1.08	0.50	6.38	727	651	98.5	83.1
0.5	.....	.....	6.29	723	650	99.1	89.1
0.1	0.6	2.5	5.62	650	579	98.2	85.71
....	.....	2.0	5.51	644	514	98.05	87.08
0.31	1.41	3.9	5.73	628	569	93.3	84.7
....	.....	2.5	5.21	610	544	98.2	87.7
0.05	.....	4.0	5.11	608	540	99.5	88.4
1.1	1.3	8.6	5.05	586	521	96.7	86.1
0.5	3.8	2.2	6.00	702	635	100.3	90.7
....	4.6	9.3	6.80	782	704	100.2	90.2
0.6	.....	2.3	6.22	714	659	96.1	91.2
0.21	.....	1.76	6.04	705	646	100.1	91.7
....	.....	6.0	4.85	579	532	98.9	90.9
....	2.0	2.0	5.06	620	524	102.3	86.4
....	0.9	.....	9.78	1085	994	100.7	92.2
....	.....	.....	8.68	967	875	100.0	90.4
0.30	0.29	2.80	9.13	989	888	97.6	87.7
0.55	0.30	3.42	9.04	980	880	97.3	87.6
0.35	.....	2.98	9.03	986	885	98.3	88.2
0.39	.....	3.41	9.06	982	882	97.6	87.7
0.42	0.26	3.02	9.06	982	883	97.6	87.8
0.35	.....	3.53	9.00	975	876	97.5	87.6
0.34	2.6	3.61	9.46	974	876	93.1	83.9
....	3.87	0.71	2.33	315	289	94.6	86.8
....	3.50	1.30	2.28	314	287	95.8	87.5
....	2.71	4.06	2.26	310	284	96.3	87.1
....	4.25	8.75	2.10	295	265	92.2	82.8
....	3.4	58.2	1.24	160	150	75.0	67.0
0.8	6.0	55.6	1.18	153	145	70.2	66.5
0.54	5.3	60.13	1.02	135	127	66.8	62.8
0.40	4.6	64.9	0.90	118	111	62.1	58.3
0.40	5.6	55.1	1.35	172	160	73.2	68.1
0.40	4.84	61.68	1.08	139	131	66.9	63.0
0.40	3.6	67.5	.91	118	113	61.2	59.2
0.10	8.2	55.5					
0.10	16.0	47.7					
....	7.0	47.0	1.32	119	115	51.4	49.6
3.0	6.0	57.0	.98	130	116	65.5	59.0
....	14.5	42.5	1.17	156	139	71.9	64.0
....	15.0	42.9	1.16	154	137	71.3	63.4
....	10.0	59.4	.81	100	99	55.3	54.8

When the analysis of gas is unknown, but the kind of gas only, the high and low limits for the various constants, based on high and low heating power, should be taken and calculations based on both. The result sought will then lie between the two limiting results.

If only a partial analysis of any one kind of gas is possible, a fair result may be obtained by finding which of the analyses given contains the elements found in about the same proportion.

The common Orsat flue-gas apparatus will determine CO, O, and CO_2 in a gas; and should this be a carbureted water-gas, the rest of the analysis can be assumed with sufficiently accurate results for engine-work by comparing with characteristic similar gas compositions. The difference between high and low values of H and $\frac{H}{a'+1}$ is due to the heat of condensation of so much steam as results from burning the hydrogen in one cubic foot of gas.

11. Fuel Characteristics.—In finding these tabulated properties needed for the thermal calculations on the gas-engine the following method of procedure has been followed. Heat per cubic foot, products hot and cold are found from values per pound of fuel given by J. Thompson, which are multiplied by the weight per cubic foot of the elementary gases, giving the results in Table VI.

TABLE VI.

Fuel.	B.T.U. per Pound.		Pounds per Cubic Foot at 62° F.	B.T.U. per Cubic Foot.	
	High.	Low.		High.	Low.
H	61524.0	51804.0	.00538	331.0	278.7
CO	4395.6	4395.6	.07498	329.58	329.58
CH_4	24021.0	21592.8	.04308	1037.22	932.38
C_2H_4	21222.0	19834.2	.07631	1619.45	1513.55

Volume of air per cubic foot gas is determined from the products and law of combination with oxygen to form those products. This volume of O found, when multiplied by 4.8 gives the air necessary on the assumption that 21% air is oxygen.

This gives

H one cubic foot requires 2.4 cu. ft. air.
CO “ “ “ “ 2.4 “ “ “
CH_4 “ “ “ “ 9.6 “ “ “
C_2H_4 “ “ “ “ 14.4 “ “ “

To illustrate the application of the above to the finding of the quantities in Table V from the gas analysis take as an example the natural gas of Table VII.

TABLE VII.

Sample Natural Gas.	Volume in 100.	B.T.U. Cubic Feet High.	B.T.U. Cubic Feet Low.	Cubic Feet Air.
H	2.18	721.58	607.57	5.23
CO	0.50	164.19	164.19	1.20
CH_4	92.60	95506.57	86338.39	888.96
C_2H_4	0.31	502.03	479.21	4.46
CO_2	0.26	0	0	
N	3.61	0	0	
O	0.34	0	0	−1.63
Total natural gas	100.00	96894.97	87589.36	898.22

The second column gives the cubic feet of each element in 100 cu. ft. gas, the third and fourth column is obtained by multiplying this by the B.T.U. per cubic foot of the elementary gas given by the previous table. The fifth column results from multiplying the second column by the air required per cubic foot elementary gas; thus for H, $2.18 \times 2.4 = 5.23$. The totals show that for one cubic foot of this natural gas, there will be developed 968.95 B.T.U. products cold or 875.89 B.T.U. products hot, and that there will be needed 8.98 cu. ft. of air to bring this about.

Hence the ratio $\frac{H}{a'+1}$ becomes $\frac{968.95}{9.98} = 97.09$ high and $\frac{875.89}{9.98} = 87.78$ low. In this way the rest of the values given in table V were obtained.

12. Pressure Ratios Observed and Calculated.—Pressure ratios observed on indicator cards will bear to the corresponding pressure ratio for the standard reference diagram a fairly constant relation. A few values are here presented. Means for finding these theoretic ratios have been given by substitution of the tabular quantities in the formula

$$\frac{P_c}{P_b} = 1 + \frac{2.2}{P_b}\left(\frac{1}{\left(\frac{P_a}{P_b}\right)^{.71}} - 1\right)\frac{H}{(a'+1)}. \quad . \quad . \quad . \quad . \quad (28)$$

As only the low heat value is liberated in the cylinder all comparisons should be made on this basis.

Gas Used.	Compression, Pounds Square Inch Absolute.	Observed Pressure Ratio, Air-card Pressure Ratio.
Blast-gas	184	56%
Mond gas	120	51%
" "	110	58%
Carbureted water-gas	90	58%
" "	60	59%
Coal-gas	60	60%
"	100	52%
Natural	115	61%
Gasolene vapor	75	56%
" "	40	58%
Gasolene in carburetor	75	40%
Kerosene vapor	65	60%
" injected	50	40%

It will be found in general that the ratio of pressures on the card will vary from 50–65% of air-card ratios when ignition is right, mixture with no excess of air or gas, no back pressure, no suction throttling, and the engine fairly free from leaks. To find the air card ratio the B.T.U. per cubic foot gas and the cubic foot air required must be known with the compression pressure. It must be noted in connection with these pressure ratios that when the indicator-card is inclined on top, as in Fig. 8, no pressure ratio for increase at constant volume can be obtained. The same is true of cards with wave tops like Fig. 9. In the latter case approximate ratios can be found by drawing a mean curve midway of crests and hollows. No extension of the expansion curve *A* in Fig. 10 can give the result. For if the pressure rise had occurred at constant volume, giving a true vertical explosion line, or nearly so, the expansion line would have fallen below as in Fig. 10, curve *B*, provided the different lines result from mixture or ignition changes and not from throttling effects. With a combustion line but slightly inclined from the vertical, as in Fig. 10, curve *B*, the pressure ratio measured at 1/10 or 2/10 stroke is usually a fair value.

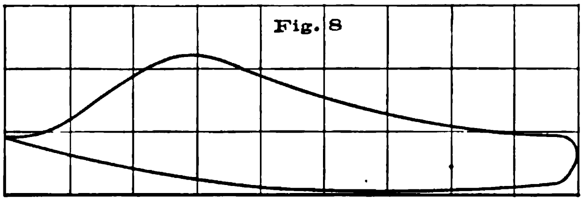


Fig. 8

The rest of the diagram, that is, the expansion curve, can easily be drawn, as the pressure ratio is the same for all points. This means that the ratio of pressures measured on the same ordinate from zero to compression and expansion curves is constant when the equations are both the same, as in this ideal case, Fig. 6.

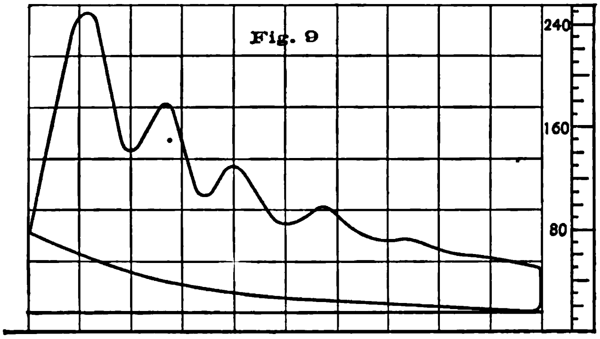


Fig. 9

$$\frac{P_c}{P_b}=\frac{P_b}{P_d}=\frac{P_y}{P_x}. \quad \dots\dots\dots \quad (29)$$

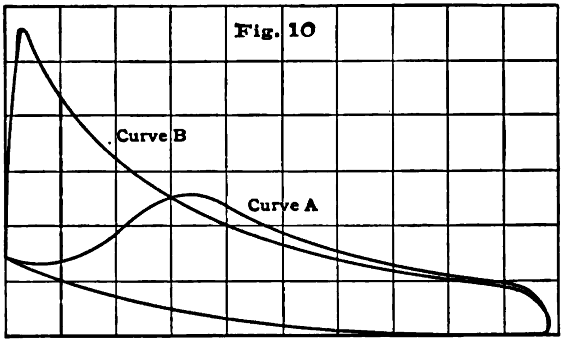


Fig. 10

Take, for example, a coal-gas compressed to four atmospheres, or 58.8 lbs. sq. in. absolute.

The final compression volume is given by

$$V_b=13.13\left(\frac{1}{4}\right)^{.71}=4.91 \text{ from the tables.}$$

The pressure increase for at 5½ to 1 mixture will be at this compression.

$$P_c - P_b = 2.2\left(\frac{1}{.374} - 1\right) \times \frac{550}{6.5} = 310 \text{ (approximately) lbs. sq. in.,}$$

the pressure ratio being 5.3 approximately for every point on expansion and compression line lying on the same vertical.

The desirability of construction of the ideal reference diagram lies in the necessity for calculating the probable forces acting on the mechanism for all points of the stroke and the variations with conditions. This will be taken up and applied later. At present, for power calculations, it is not actual forces acting behind the piston that are needed but the mean force per stroke, which can be found only after the reference diagram is established. Having done this, the main issue of mean effective pressures may be taken up for, first, the ideal diagram, and next, the fraction of this usually realized in practice.

13. **Mean Effective Pressure by Reference Diagram.**—It is well known that the efficiency of this ideal Otto cycle or standard reference air diagram is given by

$$\mathfrak{E} = 1 - \left(\frac{V_b}{V_a}\right)^{n-1} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad (30)$$

$$= 1 - \left(\frac{P_a}{P_b}\right)^{\frac{n-1}{n}} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad (31)$$

$$= \frac{W}{Q} = \frac{\text{Work done}}{\text{Heat supplied}} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (32)$$

The work done is also given by the equation

$$W' = (\text{M.E.P.})' \times (V_a - V_b) \text{ foot-pounds,}$$

where (M.E.P.)′ is pounds per square foot. Expressing M.E.P. in pounds per square inch and work in B.T.U. to conform to the expression for heat supplied,

$$W'' = \frac{144}{778}\text{M.E.P.}(V_a - V_b);$$

hence

$$\text{M.E.P.}=\frac{5.4Q}{(V_a-V_b)}\left(1-\left(\frac{P_a}{P_b}\right)^{\frac{n-1}{n}}\right). \quad . \quad . \quad . \quad . \quad (33)$$

But $$V_a-V_b=V_a\left(1-\frac{V_b}{V_a}\right)=V_a\left(1-\left(\frac{P_a}{P_b}\right)^{\frac{1}{n}}\right).$$

If now the mixture under treatment be one pound, then the following relations hold: $\frac{Q}{V_a}$ = B.T.U. per cubic foot of mixture at atmospheric pressure, and from formula (23)

$$\frac{Q}{V_a}=\frac{H}{n+a'+1}.$$

Also, V_a-V_b = displacement per pound of working gases. If the neutral comes from the clearance and has a temperature equal to T_a, the amount is given by formula (26). Giving V_b its value,

$$n'=\frac{V_b}{V_a-V_b}(a'+1)=\frac{a'+1}{\frac{V_a}{V_b}-1}=\frac{a'+1}{\left(\frac{P_b}{P_a}\right)^{\frac{1}{n}}-1}.$$

Substituting in $\frac{Q}{V_a}$, there results

$$\frac{Q}{V_a}=\frac{H}{a'+1}\,\frac{\left(\frac{P_b}{P_a}\right)^{\frac{1}{n}}-1}{\left(\frac{P_b}{P_a}\right)^{\frac{1}{n}}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

Substitution of this in equation (33), there results for mean effective pressure by the reference diagram

$$\text{M.E.P.}=5.4\frac{H}{a'+1}\left[1-\left(\frac{P_a}{P_b}\right)^{\frac{n-1}{n}}\right] \quad . \quad . \quad . \quad . \quad (35)$$

$$=5.4\frac{H}{a'+1}\left[1-\left(\frac{P_a}{P_b}\right)^{.29}\right]. \quad . \quad . \quad . \quad . \quad (36)$$

The quantity in brackets is a function of compression only, and calling it C when multiplied by 5.4,

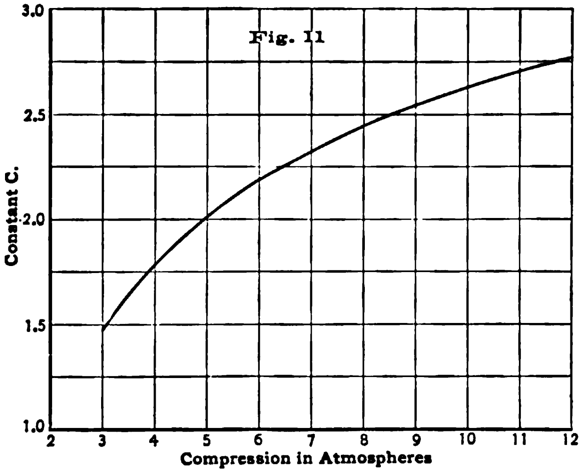
$$\text{M.E.P.} = C\frac{H}{a'+1}.$$

To facilitate calculation a table of C values is given in Table VIII.

TABLE VIII.

Compression in		$\left(\frac{P_a}{P_b}\right)^{.29}$	Mean Effective Pressure Factor $C = 5.41\left[1-\left(\frac{P_a}{P_b}\right)^{.29}\right]$
Atmospheres.	Pounds per Square Inch Absolute.		
3	44.1	.727	1.474
4	58.8	.669	1.787
5	73.5	.627	2.014
6	88.2	.595	2.187
7	102.9	.569	2.327
8	117.6	.547	2.446
9	132.3	.527	2.554
10	147.0	.513	2.630
11	161.7	.498	2.711
12	176.4	.486	2.775

These constants are plotted in a curve, Fig. 11, from which the air-card constant can be read for any compression.



14. Practicable Compression Pressures.—During compression gases become hot, and when enclosed in cylinders with walls cooled by water and with valves, igniters, and pistons all unequally warm, just how hot the gases will get will depend on the conditions of each special case. But, in any case, if the compression be carried high enough the mass will ignite itself. It is not necessary for such an explosion that the entire mass be at the temperature of ignition: one point raised to this temperature will suffice. An uncooled part, such as the center of the piston-head, may become hot enough to cause ignition at much lower pressures than would be necessary for the temperature of the rest of the walls, so that the limiting compression pressure depends for any given gas on the temperature of the hottest part of the metal in contact with the gas.

Different mixtures of any one gas, and the same mixture of different gases, permit of different compressions without this self-ignition, or preignition as it is best called. Very weak gases, that is, gases high in neutral, may be compressed higher than rich ones; thus furnace-gases may be compressed easily over one hundred and fifty pounds, while kerosene may never reach one hundred. Gases rich in hydrogen generally preignite easily; approximately one atmosphere should be deducted from the compression allowable with no hydrogen, for each 5% of hydrogen present.

The initial temperature of the air-gas-neutral mixture will have a most direct influence on the allowable compression; hence engines that scavenge with cold air either by design or by the action of a hit-or-miss governor on light loads permit of higher compressions than engines retaining always hot burnt gases in the clearance. High compression permits of the easy burning of gases, so diluted with neutral that they will not burn in the open air, and thus makes it possible to use very weak gases as fuel for engines. High compression before ignition will give more work with the same fuel, hence greater economy; similarly the mean effective pressure will be increased so that from the point of economy and power the higher the compression the better.

Recognition of this has brought forth many devices to permit of increased compression. Water injected into the cylinder either separately or mixed with the fuel, such as alcohol for example, has proved successful. Inward-projecting water-tubes, deliberate addition of excess air or neutral burnt gases or steam, have also been used, and all have been successful when properly applied, which is sometimes difficult. Such devices will be termed special, and with them the present work is not concerned. The legitimate method for increased compression in standard apparatus is to water-jacket thoroughly all parts in contact with the gases, such as piston-heads, ignitor-points, and valve-faces. All pockets that might harbor burnt gas must be eliminated as well. This is practiced only in large engines, however, in which also the weaker gases are used.

Ignition must be varied with speed, the faster the speed the earlier in the stroke should ignition take place; hence increase of speed is another legitimate means of increasing the allowable compression. In a certain kerosene-engine it was found practicable to increase compression from 55 lbs. to 85 lbs. by changing speed from 200 to 450 R.P.M. By air dilution from a compressor after suction had ceased, the compression on coal-gas was by Mr. Clerk raised from 145 to 215 lbs. per square inch. By using dilute alcohol, compression can be raised from 60 to 115 lbs. Many similar examples can be given for special cases, but these do not represent average practice.

Average practice will lie between limits which are indicated for the different fuels, and some of the conditions of working which are often contradictory will be indicated also.

Gasolene.—The compression in automobile engines varies from 45 to 95 lbs., with average practice at about 65 lbs. This is by the use of carburetors which cool the charge and not very cold circulating water, but with high speeds. These figures are in pounds per square inch above atmosphere. In engines of the slower-speed stationary class, with however more efficient cooling, though with not such simple breech ends, the compression will vary from 60 to 85 lbs., with the average about 70 lbs. In these latter engines the gasolene is usually prevaporized or used in a carburetor or spray requiring very little vacuum.

These engines are all too small to use water-cooled pistons and valves. Very often in these small engines the compression will be kept down to permit of easy turning on starting, but nothing under 40 lbs. is used.

Kerosene.—When used in hot-bulb injection and ignition-engines compression will vary from 30 to 75 lbs., depending chiefly on the speed, which should vary from 250 R.P.M. to 500 R.P.M.

for the small engines in which it is used. When previously vaporized in a device not requiring a vacuum the compression should vary from 45 to 85 lbs., the speed having an influence, the average being 65 lbs. Comparatively few engines are built for this class, as the vaporizers are few and troublesome.

Natural Gas.—Natural gas is important chiefly in larger engines than those working on gasolene and kerosene, and compressions from 75 to 130 lbs., depending on speed and the efficiency of the cooling of the valves and piston, are used, the average being about 115 lbs. for the medium and large engines using this fuel.

City Gas.—City gas will be either a coal-gas or carbureted water-gas, generally the latter. For the engines ordinarily used the compression will vary from 60 to 100 lbs., with the average at 80 lbs. These engines are seldom large, as it is not economical to generate over 75 H.P. in most cases from city gas, and being small they are not water cooled on piston and valve faces.

Producer-gas.—Producer-gas varies considerably in composition, heating power, and hydrogen, so the limits of compression are fairly wide, 100 to 160 lbs., with gases containing from 5% to 18% hydrogen, and those with water-cooled piston and valves higher. Most producer-gas engines are large, and all the latest types are being fitted with such jackets. This will probably raise these limits as well as the mean, which is now about 135 lbs.

Blast-gas.—Waste gases from blast-furnaces vary in composition and heating power as do the producer-gases, but contain but little hydrogen. In fact, the blast-furnace is but a special form of producer. The gas is weak from excessive dilution by N and CO_2, and suffers compression from 120 to 190 lbs., with the average at 155 lbs. Practically all blast-gas engines are large, developing in the neighborhood of 1000 H.P., and have water-cooled surfaces to pistons and valves when not two-cycle and valveless, and they are all of slow speed, not over 100 R.P.M.

15. Observed Mean Effective Pressure.—The following short list of some observed mean effective pressures gives in each case the average high values for the method of working. There are many ways in which the mean effective pressure of an engine can be lowered and only one best set of conditions that will produce the highest mean pressure. As these pressures are reported by different persons, there is no assurance that all are taken under the best circumstances; in fact it is certain they were not. Some of these detrimental conditions will be noticed later. It should be noted also that abnormally high mean pressures are obtained at times and cannot be continued. These values are selected at random and are not necessarily representative, but were all determined by test.

MEAN EFFECTIVE PRESSURES ON KEROSENE.

Absolute Compression.	Observed M.E.P.
46	40
63	69
65	68
68	40
70	72
50	35
55	85

MEAN EFFECTIVE PRESSURES ON GASOLENE.

Absolute Compression.	Observed M.E.P.
66	106
70	75
75	100
86	70
86	72
95	60

MEAN EFFECTIVE PRESSURES ON NATURAL GAS.

Absolute Compression.	Observed M.E.P.
115	..
127	68
130	82
135	90

MEAN EFFECTIVE PRESSURES ON CITY GAS.

Absolute Compression.	Observed M.E.P.
45	45
60	80
66	95
91	90
70	60

MEAN EFFECTIVE PRESSURES ON PRODUCER-GAS.

Absolute Compression.	Observed M.E.P.
103 (115)	63
108	88
125	51
141	83
170	73
95	100
95	90
90	62
115	103

MEAN EFFECTIVE PRESSURES ON BLAST-GAS.

Absolute Compression.	Observed M.E.P.
. . .	60
140	47
140	. .
155	81
184	. .

It is believed that the presentation of these figures will demonstrate most conclusively the difficulties surrounding the proper estimate of probable mean effective pressure and the care and experience necessary for success.

16. Diagram Factors.—Wherever it is known, the heating power of the gas and its necessary air permit of the calculation of the effective pressure of the reference air-card for an engine. This compared with the real, observed, mean effective pressure gives the diagram factor. The diagram factor then represents the percentage of work possible by the standard reference air-card that was produced by the engine. It is also a measure of the efficiency of the engine on the basis of the air-card engine as unity.

The mean effective pressure of the air-card continuously increases with the compression but in practice this result is not attained, or at least the rate of M.E.P. increase with compression becomes less as the compression increases.

Comparisons of great numbers of cards with the air-card have shown that in general the diagram factor decreases with compression.

This does not seem to be so, however, for kerosene previously vaporized, though for what reason is not known. The following values represent *fair* averages:

Diagram factors for kerosene lie between 30% to 40% for compression varying from 45 to 75 lbs. when vaporized previously, but when injected on hot bulb may run as low as 20%.

Diagram factors for gasolene lie between 25% to 40% when used in carburetors requiring a vacuum, depending on the extent of this vacuum. Gasolene 50% to 30% for compressions varying from 80 to 130 lbs., and used with but little initial vacuum.

Diagram factors for producer-gas lie between 56% to 40% for compression varying from 100 to 160 lbs.

Diagram factors for coal-gas lie between about 45% average for compression, about 80 lbs. mean.

Diagram factors for carbureted gas from lighting-mains are about 45% average for compressions varying from 60 to 90 lbs.

Diagram factors for blast-gas on compressions varying from 130 to 180 lbs. vary from 48% and 30%.

Diagram factors for natural gas on compression from 90 to 140 lbs. lie between 52% to 40%.

The diagram factors for two-cycle power cylinders may be taken as 0.8, that for four-cycle cylinders.

In selecting the factor to be applied in any particular case the designer must use judgment. There is in fact no designing possible no matter how perfect the theory nor how complete the tabular results of tests, that does not involve this element of personal experience with its resultant ripe judgment.

17. Causes of Lowered M.E.P.—A reference to the indicator cards presented later will show variations of form and area resulting from different conditions, and it is by reason of these conditions as evidenced by the cards that mean effective pressures are lowered from the best possible with the compression and fuel to nothing at all. These detrimental causes are of three classes:

Class 1. Those that change the weight of the cylinder charge.

Class 2. Improper treatment of full charges.

Class 3. Improper treatment of partial charges.

FIRST CLASS.

The weight of charge may be lessened by:

Too small an opening in the suction passages acting throughout the whole or a part of the stroke.

Improperly set valves.

Too heavy a spring on an automatic suction-valve.

Too small a valve diameter.

Too small a valve-lift.

Very crooked passages.

Governor action.

Too much back pressure-preventing entrance of new mixture.

Sudden increase of back pressure due to exhaust of one cylinder which is just beginning to exhaust, blowing back into another cylinder whose exhaust is just closing.

Heating of incoming gases by burnt gases in clearance or hot passages.

Too high a speed.

Improper capacity of suction-chamber in two-cycle engines.

Improper cam curves or proper cam curves improperly set.

SECOND CLASS.

Leaks.

Improper mixture, giving a slower rate of propagation for the same piston speed, making a flat-top diagram.

Proper mixture with too high a piston speed.

Too early ignition.

Too late ignition.

Improper mixture and ignition together.

Improper release of gases when too early.

Not the best compression pressure.

Poor form of clearance or breech-end.

THIRD CLASS.

This class by the double action of the first two will produce results doubly intense, and mixtures may be too slow burning, ignition wrong, and only partial charges present; so with all these actions the mean effective pressure can be only very low.

18. Mean Effective Pressure and Horse-power Formulas.—Either through ignorance of the various influences or conditions that affect the mean effective pressures or by giving the effect of these influences a numerical value for some particular case, formulas have been proposed for the horse-power of gas-engines that are quite simple in form. Some of these formulas are quite legitimate, but some cannot be justified. Some of these will be set down and compared.

Frederic Grover has proposed the formula

$$P = 2C - .01C^2, \quad \ldots\ldots \quad (37)$$

where C is the compression pressure in pounds per square inch above atmosphere.

This formula neglects absolutely any of the fuel characteristics and therefore can be applied only to one fuel rationally with the constants given. A change of constants would be necessary for adaptation to another fuel.

The following mean effective pressures result from the formula for the compressions given:

Compression above Atmosphere.	(P) by Grover Formula.
40	64
60	84
80	106
100	100
120	96
140	84

This formula expresses the fact that on too high a compression the mean effective pressure will fall off, but expresses the result too arbitrarily to be trusted; it might, however, be adapted as suggested by change of constants to fit conditions.

A formula for I.H.P. as given by several writers is

$$\text{I.H.P.} = \frac{d^2 \times (l+c) \times \left(\left(\frac{l+c}{c} \right)^{\frac{1}{3}} - 1 \right) \times N}{9160}; \quad \ldots \quad (38)$$

when d = diameter piston in inches,
l = stroke in inches,
c = clearance volume ÷ piston area in inches,
N = R.P.M.,
$\frac{l+c}{c}=\frac{\text{total cylinder volume}}{\text{clearance volume}}$.

Rewriting and equating with the regular formula for four-cycle single-acting cylinder,

$$\text{I.H.P.}=\frac{d^2l\left(1+\frac{c}{l}\right)\left(\left(\frac{l}{c}+1\right)^{\frac{1}{3}}-1\right)N}{9160}=\frac{1}{4}\frac{PL_aN}{33000}.$$

Here $l=12L$, $d^2=\frac{4a}{\pi}$; hence, calling

$$\frac{l+c}{c}=\frac{V_a}{V_b},\qquad \frac{l+c}{l}=\frac{V_a}{V_a-V_b},$$

$$\therefore\ \frac{\frac{V_a}{V_a-V_b}\left(\left(\frac{V_a}{V_b}\right)^{\frac{1}{3}}-1\right)\times 12\times 4}{9160\times\pi}=\frac{1}{4}\times\frac{P}{33000};$$

or putting all the constants in one C',

$$P=C'\left(\frac{V_a}{V_a-V_b}\right)\left(\left(\frac{V_a}{V_b}\right)^{\frac{1}{3}}-1\right)$$

$$=C'\frac{V_a}{V_a-V_b}\frac{P_b}{P_a}\left[1-\left(\frac{P_a}{P_b}\right)^{\frac{n-1}{n}}\right],$$

if the expansion and compression lines had the exponent $1.33=n$, which, when compared with equation (35), will be found to be the same with the exception of the term containing the fuel characteristics.

Thus this formula is incomplete as it stands, but by proper constants may be made to fit the case of any fuel for a limited range.

The larger number of formulas have the form

$$\text{B.H.P.}=Md^2LM, \quad \ldots\ldots \quad (39)$$

where M is a constant, values of which are proposed by a good many engineers here and abroad. The formula as it stands involves three assumptions, none of them true:

1°. That mechanical efficiency or the ratio of indicated to brake horse-power is always the same.

2°. That P is independent of compression.

3°. That P is the same for all fuels.

By having a set of constants for each fuel the case is better, but not much. Such formulas, while easy to remember, are apt to deceive by making the case seem simpler than it is, and by giving results that sometimes happen to come right, thus giving confidence to the designer only to lead him to failure in another case with different conditions.

19. Cylinder Diameter.—The first step in determining the cylinder diameter for a given power is to decide on the number of cylinders, and in many cases, in fact in most cases, this will depend largely on questions of uniformity of turning effort and balance needed, as this calls for multiplication of cylinders, while cost of building demands as few cylinders as possible. It will be well to find the diameter and stroke for one, two, three, four, etc., as may be required and select the dimensions and number after the question of balance and turning effort has been decided.

Formula (8) gave for four-cycle engines

$$\text{I.H.P.} = \frac{1}{4} \frac{PaS}{33000}.$$

Putting

$$a = \frac{\pi d^2}{4} = .7854 d^2,$$

$$\text{I.H.P.} = \frac{.7854 P d^2 S}{4 \times 33000};$$

from which

$$d = \sqrt{\frac{4 \times 33000}{.7854} \times \frac{\text{I.H.P.}}{P \times S}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (40)$$

for a single four-cycle cylinder, and for a two-cycle cylinder

$$d = \sqrt{\frac{2 \times 33000}{7854} \times \frac{\text{I.H.P.}}{P \times S}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (41)$$

This will give a diameter of cylinder for the indicated horsepower required when mean effective pressure and piston speed are determined.

20. Overload Factor.—In a gas-engine there is no means of carrying more load than the maximum resulting from its dimensions and the mean pressures due to the combustion. By making the cylinder a size larger than is necessary to carry the rated or normal load, an overload can be carried. If the overload to be provided for is $\alpha\%$ of the normal I.H.P. the engine will give $\left(1+\frac{\alpha}{100}\right)$ I.H.P. and the diameter must be increased to $\sqrt{\left(1+\frac{\alpha}{100}\right)}$ times the no-overload diameter.

Let $\alpha = \%$ overload to be provided for, then for four-cycle cylinder

$$d=\sqrt{\frac{4\times 33000}{.7854}\times\frac{\left(1+\frac{\alpha}{100}\right)\text{I.H.P.}}{P\times S}}, \quad . \quad . \quad . \quad . \quad (42)$$

and for a two-cycle cylinder

$$d=\sqrt{\frac{2\times 33000}{.7854}\times\frac{\left(1\times\frac{\alpha}{100}\right)\text{I.H.P.}}{P\times S}}. \quad . \quad . \quad . \quad (43)$$

The nearest reasonable dimension to d should be chosen so as to give an even number of sixty-fourths or hundredths.

21. Stroke.—The stroke is given by the relation between piston speed and R.P.M., but the stroke thus determined may be anything, as R.P.M. varies and must be checked by the ratios of stroke to diameter usually practiced.

This ratio of $\frac{l}{d}$ will always be between 1.0 and 2.0, and it makes but little difference just what value in these limits is used except for questions of convenience.

For automobile engines the ratio will vary from 1.0 to 1.3.

Small stationary engines usually have a little larger ratio, say from 1.2 to 2.00, while very large engines use ratios lying anywhere within the limits.

The exact value of the ratio will, of course, be such as to give a convenient number of inches and fractions to both stroke and diameter.

Some dimensions of engines already built, selected at random, will illustrate this point:

Size.	Ratio $\frac{l}{d}$.	Number of Cylinders.	R.P.M.
4½×5½	1.2	4 single-acting four-cycle	900
6×9	1.5	2 " "	250
5¾×12½	1.6	1 " "	300
11½×21	1.7	1 " "	170
20.08×25.56	1.3	1 double-acting four-cycle	150
25×25	1.0	1 single-acting four-cycle	180
29×40	1.4	1 " "	110
40.55×51.24	1.3	1 double-acting four-cycle	90

22. **Performance, Economy, and Thermal Efficiency.**—The per cent of heat supplied to an engine that is transformed into work is the thermal efficiency.

Heat supplied to an engine is measured in B.T.U. per hour, and is the product of the weight or volume of fuel per hour and the calorific power of the fuel in B.T.U. per pound or B.T.U. per cubic foot respectively. This calorific power of the fuel may or may not be taken as that given by the calorimeter. When the fuel contains any hydrogen its combustion yields steam and the condensation of the steam yields water. This latter condensation adds to the true heat of combustion the latent heat of so much of the steam as is condensed. Engines can never use this heat of condensation. Wherever the hydrogen in the fuel is known, two calorific values can be found, and these are called the higher and the lower values. The former is that given by the calorimeter, and includes the heat of steam condensation; the latter excludes this latent heat.

Work done is measured in either B.H.P. or I.H.P.; that is, foot-pounds per minute at the wheel rim divided by 33,000, or foot-pounds per minute behind the piston divided by 33,000. The latter is greater than the former by so much H.P. as is lost in the engine itself, as was explained under Mechanical Efficiency.

The performance of an engine can rationally be given in several ways, but without defining which quantities are compared, the terms economy or thermal efficiency may be misleading, especially when comparing two engines whose thermal efficiencies are based on different units.

Thermal efficiency may be expressed as follows:

$$\text{(1)}\ E_1=\frac{\text{B.H.P.}\times 33000}{\text{B.T.U. per minute in fuel (higher value)}\times 778}$$

$$=\frac{\text{ft.-lbs. per minute at brake}}{\text{ft.-lbs. per minute in fuel (higher value)}}. \quad . \quad . \quad . \quad . \quad (44)$$

This may be called the thermal efficiency based on brake horsepower and the higher fuel value.

(2) Similarly thermal efficiency based on brake horse-power and lower fuel value is equal to

$$E_2=\frac{\text{B.H.P.}\times 33000}{\text{B.T.U. per minute in fuel (lower value)}\times 778}. \quad . \quad (45)$$

(3) Using indicated horse-power as the output unit the following two values will be given:

$$E_3=\frac{\text{I.H.P.}\times 33000}{\text{B.T.U. per minute in fuel (higher value)}\times 778} \quad . \quad (46)$$

and

$$\text{(4)}\ E_4=\frac{\text{I.H.P.}\times 33000}{\text{B.T.U. per minute in fuel (lower value)}\times 778}. \quad . \quad (47)$$

All these ratios are rational and justifiable, but it must be carefully defined, in expressing thermal efficiencies, which of these quantities is meant.

Suppose an engine gave 4 B.H.P. and 5 I.H.P. with 100 cu. ft. of gas per hour whose higher calorific value was 600 and whose lower was 550 B.T.U. per cubic foot, then we should have

$$E_1=\frac{4\times 33000}{\frac{100}{60}\times 600\times 778}=17\%\ \text{approximately}; \quad . \quad . \quad (48)$$

$$E_2=\frac{4\times 33000}{\frac{100}{60}\times 550\times 778}=18+\%\quad \text{“} \quad . \quad . \quad (49)$$

$$E_3=\frac{5\times 33000}{\frac{100}{60}\times 600\times 778}=21\% \text{ approximately}; \quad . \ . \quad (50)$$

$$E_4=\frac{5\times 33000}{\frac{100}{60}\times 550\times 778}=23\% \quad \text{"} \quad . \ . \quad (51)$$

It will appear from this what confusion will result from neglect of the accurate definition of efficiency arising from the different and all correct methods of defining input and output of energy. In most cases the higher and lower calorific values do not differ so much as here assumed. In what follows these definitions will always be given.

Economy is the term applied to the amount of energy per hour that must be supplied to produce one horse-power, and the following expressions for it are all correct.

Economy is seldom expressed by symbols, but is always written out as follows:

B.T.U. per hour per I.H.P., lower calorific value.
" " " " " higher " "
" " " " B.H.P., lower " "
" " " " " higher " "

The economy of gas-engines will not vary greatly when comparisons are made on the best conditions in each case and an easy number to remember is 10000 B.T.U. (lower value) per I.H.P. hour as the average performance. It must not be imagined that no variations occur from this, for nothing could be farther from the truth, as will be shown presently.

23. **Efficiency by Reference Diagram.**—It is desirable in studying efficiencies and economies to have a standard of reference, and actual results will be easily remembered as some definite fraction, 60% for example, of what would be the standard. For this purpose the standard reference diagram already developed is useful, as it expresses what is the best an engine could do with this *cycle* or *method* of *working* and what any exploding engine would do under the best conditions with no losses. Any claim of better results in

performance by the misinformed, exceeding that possible by the standard reference diagram, can be immediately checked.

As this standard reference diagram is based on certain operations on the gas itself it gives the possibilities of the method excluding mechanisms, so that all efficiencies or economies must be expressed in terms of I.H.P. instead of B.H.P. output.

As the gases are confessedly discharged at a temperature higher than the condensation of steam at the pressure which is developed, the lower calorific value must be used. For the standard reference diagram the efficiency depends only on the extent of the compression and calling all efficiencies of this diagram by the letter $\mathfrak{E}$, using the same subscripts.

$$\mathfrak{E}_4 = 1 - \left(\frac{V_b}{V_a}\right)^{.4} \text{ in terms of volumes, } \quad . \quad . \quad . \quad . \quad (52)$$

$$\mathfrak{E}_4 = 1 - \left(\frac{P_a}{P_b}\right)^{.29} \text{ in terms of pressures; } \quad . \quad . \quad . \quad . \quad (53)$$

where P_b = absolute compression pressure,
P_a = atmospheric pressure.

A table of these efficiencies is added for reference, Table IX, the $\mathfrak{E}_4$ referring to the ratio of foot-lbs. per minute by indicated horse-power to foot-lbs. per minute by the lower calorific value of the fuel.

TABLE IX.

EFFICIENCY $\mathfrak{E}_4$ FOR STANDARD REFERENCE DIAGRAM.

Compression in Atmospheres.	Compression, in Pounds per Square Inch, Absolute.	$\mathfrak{E}_4$.	Compression in Atmospheres.	Compression, in Pounds per Square Inch, Absolute.	$\mathfrak{E}_4$.
3.0	44.1	28%	8.0	117.6	45%
3.5	51.4	31%	8.5	124.9	46%
4.0	58.8	33%	9.0	132.3	47%
4.5	66.1	35%	9.5	139.6	48%
5.0	73.5	37%	10.0	147.0	49%
5.5	81.8	39%	10.5	154.3	49%
6.0	88.2	41%	11.0	161.7	50%
6.5	95.5	42%	11.5	169.0	50%
7.0	102.9	43%	12.0	176.4	50%
7.5	110.2	44%			

As this table is to be used for comparison with observed efficiencies, which can never be accurately known to three places of decimals, the per cent is here given to the nearest two figures only.

24. **Economy.**—If there were no losses, one B.T.U. per minute would give 778 ft.-lbs. per minute behind the piston, or 60 B.T.U. per hour would give 778 ft.-lbs. per minute behind the piston. Hence 60 B.T.U. per hour should give $\frac{778}{33000}$ I.H.P., whence B.T.U. per I.H.P.-hour $=\frac{33000}{778}\times 60=2545$, with no losses of any kind. This then is the theoretic economy for any engine. When the engine is a gas-engine and the heat derived from fuel with hydrogen, the precautions noted in determining this B.T.U. per hour from the quantity of fuel and its calorific power should be remembered and when higher or lower value is used it should be so stated.

Guarantees of performance are best stated at so many B.T.U. per hour per I.H.P. or B.H.P. as *determined* by the *Junker* calorimeter for gases or by the *Mahler* bomb for solid and liquid fuels. This is done because in the test for fulfillment this is an easily determined quantity, but gives the higher calorific value. As will be shown later, the fuel may change with conditions, especially in producer work, the method, time, load, place, number of times, etc., of measurement of hydrogen should be clearly defined as well in the contract.

It sometimes happens that in large engines the quantity of gas cannot be measured, but by analysis and calorimeters its calorific value can be found direct and the air necessary for its combustion calculated. These measurements furnish an easy means of expressing approximately its thermal efficiency or economy by using the standard reference diagram. By slightly varying the mixture there will be found a best card; that is, a card having the largest mean effective pressure. This will correspond to the *mixture* having the greatest heat of combustion which will be approximately that mixture of air to gas determined by the chemical analysis as theoretically correct.

Thus there will be at hand,

1°. Air to gas.

2°. Hydrogen.

3°. Actual P for 1° and 2°, called P', from which can be calculated,

4°. Theoretic P for standard reference diagram and the ratio of actual P' to theoretic P.

From this can be calculated the B.T.U. per hour per I.H.P. by multiplying the theoretic economy by the above ratio. One cubic foot of gas per stroke calls for a displacement of $(a'+1)$ cubic feet and yields PLa or 144 PLA foot-lbs., but LA is the displacement. Hence

$$144PLA = 144P(a'+1);$$

$$60\times H\times N \text{ B.T.U. per hour yields } \frac{144P(a'+1)N}{33000}\text{I.H.P.}$$

Hence B.T.U. per hour (low) per indicated horse-power is

$$\frac{60\times H\times N\times 33000}{144P(a'+1)N}.$$

But
$$P=\frac{778}{144}\frac{H}{a'+1}\left(1-\left(\frac{P_a}{P_b}\right)^{\frac{n-1}{n}}\right),$$

from formula (35). Hence

$$\left.\begin{array}{l}\text{B.T.U. per hour}\\ \text{(low) per I.H.P.}\end{array}\right\}=\frac{60\times H\times 33000}{144\times\frac{778}{144}\times\frac{H}{(a'+1)}\times(a'+1)\times\left(1-\left(\frac{P_a}{P_b}\right)^{\frac{n-1}{n}}\right)}$$
$$=\frac{60\times 33000}{778}\frac{1}{\left(1-\left(\frac{P_a}{P_b}\right)^{\frac{n-1}{n}}\right)}.$$

But $\frac{60\times 33000}{778}=2545=$ B.T.U. per I.H.P. per hour for any engine with no losses, while $\left(1-\left(\frac{P_a}{P_b}\right)^{\frac{n-1}{n}}\right)=\mathfrak{E}_4$, the thermal efficiency of the standard reference diagram based on I.H.P. and lower calorific value.

This gives

$$\left.\begin{array}{l}\text{B.T.U. per hour (low) per I.H.P. for}\\ \quad\text{the standard reference diagram}\end{array}\right\}=\frac{2545}{\mathfrak{E}_4}, \quad . \quad . \quad (54)$$

as might have been expected without analysis.

If now, using this fuel, the actual engine gave a mean effective pressure P', then its thermal efficiency E_4 would be given by

$$E_4 = \mathcal{E}_4 \frac{P'}{P} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (55)$$

That is, the thermal efficiency of the actual case is equal to the thermal efficiency by the standard reference diagram multiplied by the ratio of standard to actual M.E.P. This latter ratio is given for various cases in practice and has been called the diagram factor.

From the above we may write

$$\text{B.T.U. per hour (low) per I.H.P.} = \frac{\text{Constant}}{P} \quad . \quad . \qquad (56)$$

for the same conditions of mixture, gas, and speed, with different mean effective pressures. Hence

$$\left.\begin{array}{l}\text{B.T.U. per hour per} \\ \text{I.H.P. for engine with} \\ \text{mean effective pres-} \\ \text{sure } P'\end{array}\right\} = \frac{P'}{P} \times \left\{\begin{array}{l}\text{B.T.U. per hour per} \\ \text{I.H.P. for reference} \\ \text{diagram with mean} \\ \text{effective pressure } P\end{array}\right\} . \qquad (57)$$

This $\frac{P'}{P}$ is the diagram factor already discussed.

Economy in terms of B.T.U. per hour per I.H.P. for the standard reference diagram is given in Table X.

TABLE X.

ECONOMY OF STANDARD REFERENCE DIAGRAM.

In B.T.U. per hour per I.H.P. $= \frac{2545}{\mathcal{E}_4}$.

Compression in Atmospheres.	Compression, Pounds Sq. Inch, Absolute.	$\mathcal{E}_4$.	B.T.U. per Hour per I.H.P. by Standard Reference Diagram.	Compression in Atmospheres.	Compression, Pounds Sq. Inch, Absolute.	$\mathcal{E}_4$.	B.T.U. per Hour per I.H.P. by Standard Reference Diagram.
3.0	44.1	28%	9089	8.0	117.6	45%	5655
3.5	51.4	31%	8210	8.5	124.9	46%	5535
4.0	58.8	33%	7712	9.0	132.3	47%	5415
4.5	66.1	35%	7271	9.5	139.6	48%	5302
5.0	73.5	37%	6879	10.0	147.0	49%	5194
5.5	81.8	39%	6527	10.5	154.3	49%	5194
6.0	88.2	41%	6207	11.0	161.7	50%	5090
6.5	95.5	42%	6059	11.5	169.0	50%	5090
7.0	102.9	43%	5919	12.0	176.4	51%	4990
7.5	110.2	44%	5785				

It thus appears that, given the diagram factor for a given case, the thermal efficiency can be obtained at once as well as the economy, provided a table of values for the various effects by the standard reference diagram is available or a curve like Fig. 12. When not available, the quantities needed are easily calculated.

Anything that will affect the mean effective pressure of a full charge will affect both thermal efficiency and economy based on indicated horse-power. Anything that affects the engine losses in addition to the above will affect the thermal efficiency and economy based on brake horse-power.

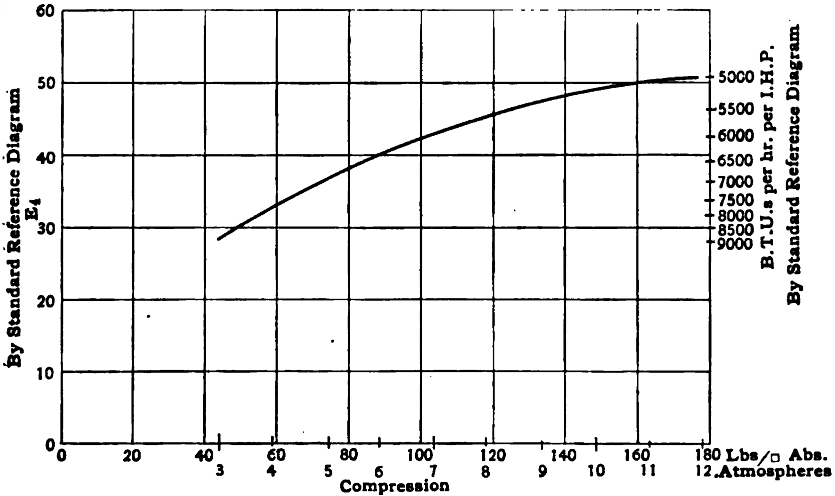


Fig. 12

25. **Application of Diagram Factors.**—Diagram factors found for mean effective pressures can be applied to economy and efficiency determination, but always with exercise of some judgment. Results from the published reports of distinguished experimenters are always a little too good to be used in the every-day routine of designing, building, and operating for two reasons. First, they are generally the best results obtainable from the engine adjusted to its best conditions, and secondly, they are obtained by the best men of the profession, who will accept and publish only the best results as they recognize them, and probably of many days' running only a few hours' results are accepted. On the other hand, the buyer will demand, and rightly, an economy guarantee based on the ordinary condition of the apparatus.

To illustrate this point of caution some results reported by Mr. Clerk in a very recent lecture will be given and compared with the figures presented for diagram factors. He calculates the air-card efficiency for the engine tests reported by Professor Meyer in Germany and Professor Burstall in England.

Observed Efficiency. E_3.	Air-card Efficiency. $\mathfrak{E}$.	$\frac{\text{Actual}}{\text{Standard}} = \frac{E_3}{\mathfrak{E}} =$ factor.	Experimenter.
.250	.44	.58	Meyer
.244	.42	.58	"
.214	.37	.58	"
.188	.33	.57	"
.189	.33	.57	Burstall
.212	.36	.59	"
.219	.43	.51	"
.231	.47	.50	"
.166	.33	.50	"
.187	.36	.52	"
.172	.43	.40	"
.181	.47	.38	"

The results of Meyer were on compressions 40 to 80, and Burstall also used variable compressions, but with compression rings, to bring about these results. The best results here for diagram factors are about 9% greater than given in the text as reliable. In general it will be found that these results of experts on engines particularly correct in adjustment will run from 5% to 15% better than the diagram factor that it is wise to employ. If the diagram factor for a particular case seems to be about right at 45% then it is probable that with everything at its best the factor may prove to be 50% to 60%.

In fact, Mr. Clerk says that actual efficiencies will vary from .5 to .6 of the air standard efficiencies, although it will be better not to depend on this but rather to go below, using figures .40 to .55 for ordinary full-charge engines and lower still for special kinds noted.

26. Variation in Performance with Conditions.—The investigation of nature of the change in economy and efficiency with some conditions most frequently encountered is desirable here, as the above discussion is based on the best results for an engine. A guarantee of economy should include conditions under which it is to be measured, full load or not, best ignition and mixture or constant setting, highest speed or mean speed, gas pressure, etc., as all these things exert influences.

A 10-I.H.P. hit-or-miss governed Otto engine on 550 (low) carbreted water-gas gave, with different settings of the gas-valve, efficiencies varying both ways from a maximum indicating on the one hand excess of gas and on the other excess of air as follows:

Gas-valve Number.	$E_1 = \frac{\text{Brake energy}}{\text{Fuel energy (higher value)}}$.
9	16.0%
8	16.5%
7	18.0%
6	19.0%
5	10.0%

The engine was in ordinary running condition, with the usual small leaks and constant ignition.

These results are plotted in Fig. 13. Humphrey reports a similar curve, Fig. 14, for Mond gas from a 400-H.P. Crossley engine, but expressed in terms of quantity of gas per stroke.

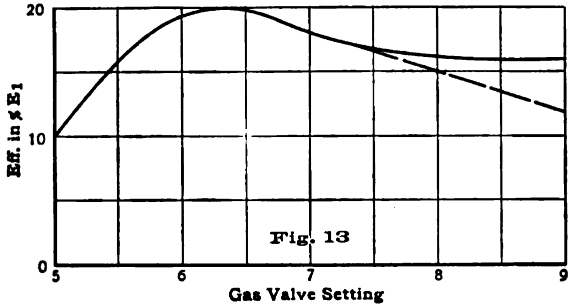
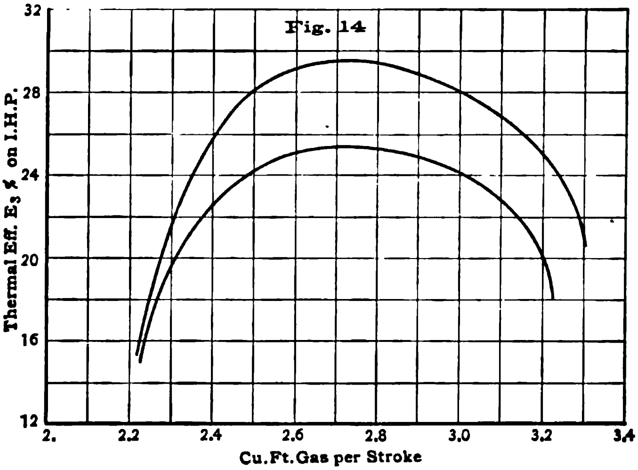


Fig. 13

When ignition and mixture were kept constant at the most advantageous point on a hit-or-miss engine of 6 B.H.P. the following results in efficiency with load were obtained:

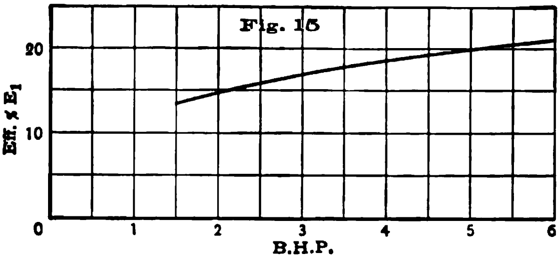
Load B.H.P.	$E_1 = \frac{\text{Brake energy}}{\text{Fuel energy (higher value)}}$.
0	
2	15.5%
4	18.5%
6	22.0%
8	22.5%

These results are plotted in Fig. 15 and show the nature of the curve, which would not change much in form for more or less leakage than existed.



A small kerosene injection engine, hot-bulb type, with the usual lack of uniformity of mixture and governed by variation of mixture, gave the following results with load:

Load B.H.P.	$E_1 = \frac{\text{Brake energy}}{\text{Fuel energy (higher value)}}$.
.75	13%
1.25	14%
2.00	15%



These results are plotted in Fig. 16 to show the nature of the curve, which appears, as it should, not very different from Fig. 15 or the first part of Fig. 13 for the conditions.

With gasolene previously vaporized and full charges Halladay and Hodge report a maximum of

$$E_1 = 14.9\% \text{ at } 9.43 \text{ B.H.P. (about } \tfrac{3}{4} \text{ load).}$$

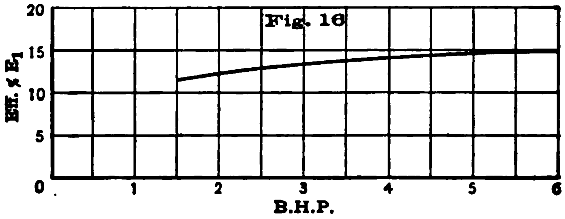


Fig. 16

Using kerosene in a similar way and in the same engine

$$E_1 = 19.2\% \text{ at } 9.1 \text{ B.H.P. (about } \tfrac{3}{4} \text{ load).}$$

These figures are lower where kerosene is used in hot bulbs by injection and where gasolene is used in carburetors requiring much vacuum.

The Society of Arts' trial reported by Dr. Kennedy gave for the Atkinson engine on gas 623 B.T.U. (high) the results tabulated.

I.H.P.	B.H.P.	Cu. Ft. Gas per B.H.P.-hour.	E_1.
11.15	9.48	18.82	21.7
6.95	4.74	19.29	21.2
2.3	0	20.50	19.9

These results are plotted in Fig. 17.

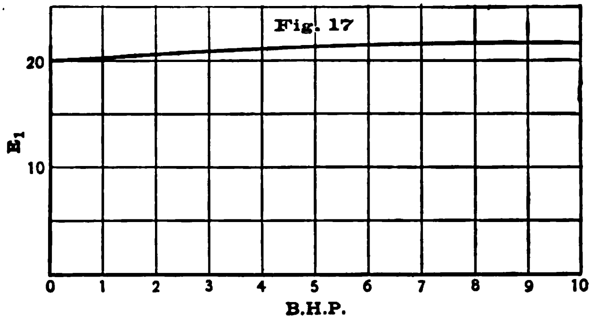


Fig. 17

There was also reported for a 17-H.P. Crossley $E_3 = 21.2\%$ (high), and for a 5-H.P. Griffin $E_5 = 19.2\%$ (high).

Tests by Aimé Witz on a Charon engine gave results as follows, plotted in Fig. 18:

B.H.P.	Cu. Ft. Gas per B.H.P.-hour.	E_1
60	15.1	27.1
53	16.5	24.8
41	20.0	20.4
30	24.5	16.7

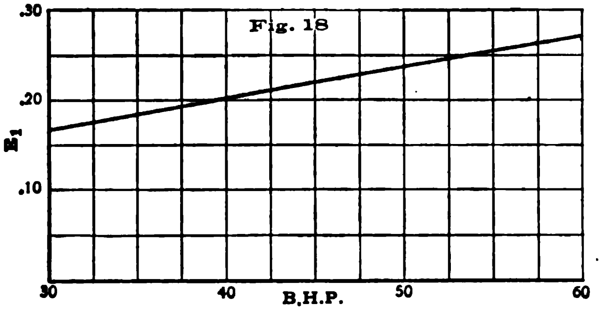


Fig. 18

Another test reported by Philip Dawson gave the results plotted in Fig. 19 and tabulated below. The curve is given more to show its nature than for comparison.

B.H.P.	Cu. Ft. Gas per B.H.P.-hour.
10	19.86
15	13.44
20	11.48
25	10.25
28	10.08

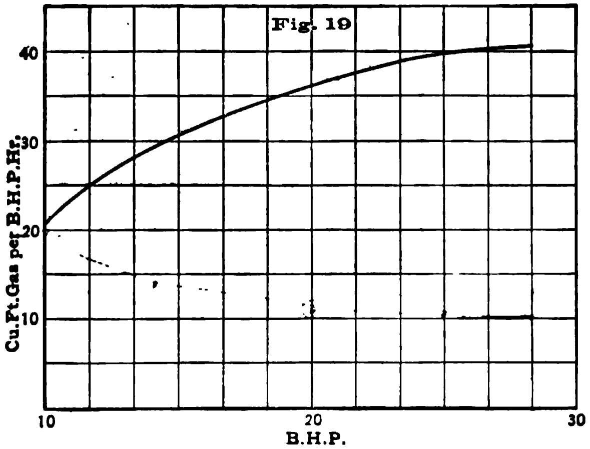


Fig. 19

A Stockport engine tested by Mathot gave the following, which show a curve nearly flat for more than half load:

Electric H.P.	Cu. Ft. Gas per E.H.P.-hour.
44.0	17.34
50.2	15.65
25.1	19.56

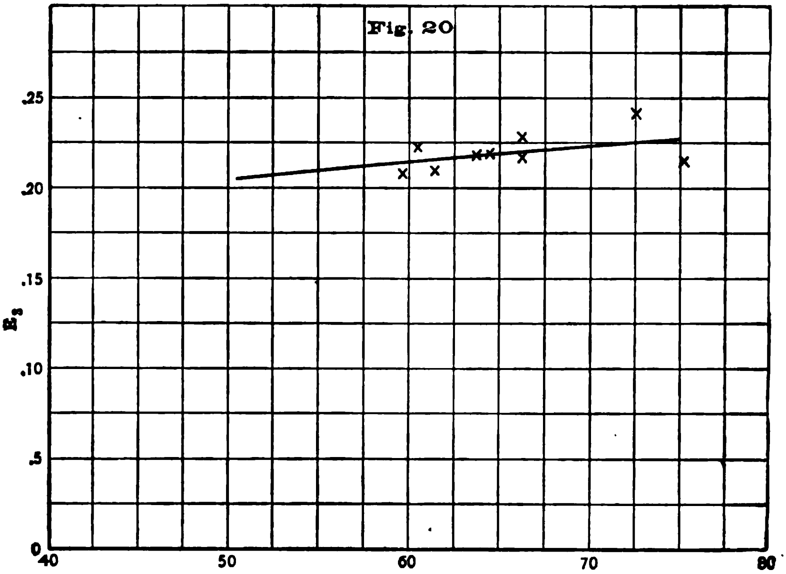


Fig. 20

A single-cylinder Crossley engine 8½×24, at Worcester, Mass., gave, on producer-gas, 140 B.T.U. per cu. ft. (high) (Fig. 20).

I.H.P.	Cu. Ft. Gas per I.H.P.-hour.	$E_3 = \frac{\text{Indicated energy}}{\text{Fuel energy (high)}}$
75.2	86.5	21.5
72.4	81.8	24.1
66.5	86.4	22.8
63.8	90.0	21.9
59.6	89.3	20.6
61.5	88.0	20.9
60.5	81.9	22.4
66.7	86.8	21.2
64.4	84.0	21.9

These results are nearly all at full load and show how, even with precaution against change of conditions, the efficiency from observations may vary considerably. The nature of the mean curve, however, is the same as for other similar cases.

Professor Robertson reports a test on a three-cylinder 13″×14″ Westinghouse engine at Lafayette on 1100 B.T.U. natural gas (high) which is reliable, though not the only one made.

I.H.P.	Cu. Ft. Gas per I.H.P.-hour.	$E_2 = \frac{\text{Indicated energy}}{\text{Fuel energy (high)}}$
111.55	13.4	18.99
109.43	14.73	17.27
102.66	14.71	17.30
92.27	11.87	21.44
79.31	12.94	19.67
39.75	17.75	14.34

Another test reported by the builders, on a 600-H.P. 25″×30″ engine gives both E_2 and E_4 as well as economy on B.H.P. and

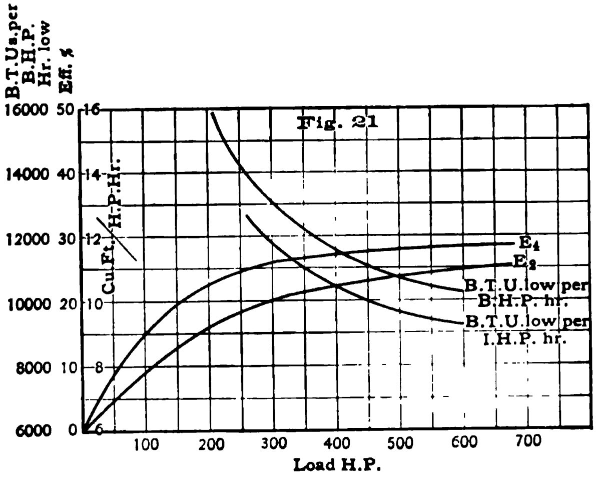


Fig. 21

I.H.P. This is plotted in Fig. 21 to show how very rapidly change in economy results with throttling effect due to light loads, rather than to give the actual figures, which are good at full load rather too good in fact for continued performance under working conditions.

A test of the producer plant and engines at Cassel by A. Witz gave some results on coal. The coal contained 10% ash and 1% water and test ran six hours, using

In producers	323	lbs. coal.
" boiler	126	" "
Total	449	" "

Total coal per hour, 74 lbs.

I.H.P.	60.08
Friction H.P.	9.12
B.H.P.	56.96

Pounds fuel per hour = 123 lbs., total on ¾-load constant but the calorific value of the coal is not given.

In a Dowson plant of Fielding and Platt the guarantee was for 85 cu. ft. of 145 B.T.U. gas per B.H.P.-hour or 12,825 B.T.U. higher value.

A common American guarantee for producer plants of 1000 H.P. more or less is 1 B.H.P. on 1 lb. coal gasified, excluding that used in the boiler, or 11,000 gas B.T.U. (high) per hour per B.H.P.

There is reported by the builders a result of 2.97 lbs. dry wood gasified in Loomis-Pettibone producers at Nacozari, Mexico, the gas being used in Crossley engines.

Mond-gas results, reported by Mr. H. A. Humphrey, show for 400-H.P. Crossley (142.5 high) $E_3 = 23.76\%$.

500-H.P. Premier (143.8 high), $E_3 = 25.6\%$, the latter engine having higher compression than the former.

Furnace-gas results, reported by C. A. Cochrane, show for a 600-H.P. Cockerell engine at Ormsby Iron Works, on 98 gas (high), the engine compressing air,

$$E_3 = 25.43\% \text{ (I.H.P.)},$$
$$E_1 = 19.32\% \text{ (Air H.P.)};$$

this is for full and best load.

Much of the gain in efficiency with blast and other weak gases is due largely to the compression, and the nature of the increase is shown by the Kennedy reports to the Society of Arts Committee on a series of changes on compression. This curve is plotted in Fig. 22.

Compression.	$E_1 = \frac{\text{Brake energy}}{\text{Fuel energy (high)}}$.
55	13.2
71	14.0
93	13.4
124	15.9

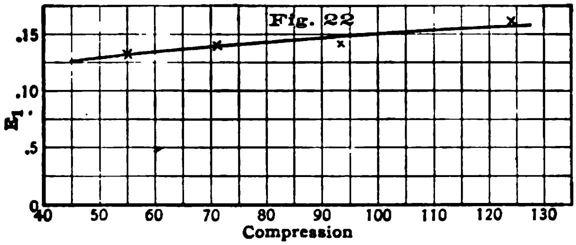


Fig. 22

The statement of an average E is not a good thing for where E varies, there is a reason to be sought in the conditions, and experimenters never give all the desirable conditions. The best that can be done to guide the designer is to give two limits between which the E will lie for the best case in any engine; that is, with full charges, best mixture, proper ignition, no leaks, no back pressure.

The economy of a small engine is not so good as for a large engine of identical design, and this should be borne in mind by the designer. To show just how the efficiency and economy vary with size, it is necessary to test a full series of sizes of the same make and of identical design. This has never been done, chiefly because in putting on the market a series of sizes, identical design is not the rule, but rather approximations are made which permit of interuse of patterns for certain parts and because of the desirability of keeping dimensions to whole numbers or simple fractions of inches.

PART II.

FORCES IN THE ENGINE DUE TO GAS PRESSURE AND INERTIA.

27. **The Actual Indicator Card.**—Many changes of form in the curves constituting the indicator card of a gas-engine occur in practice from various causes to be noted, and both the nature of the changes and their extent and variety greatly exceed those found on steam-cards. This makes interpretation of real cards difficult, and prediction of probable cards for a new engine still more so when the principles which cause the changes of form are not clear. These principles while simple in themselves are complex in application, and results cannot be too thoroughly studied, more so than the space here allowed will permit, by reference to papers and other published works on the properties of explosive mixtures.

The curve characteristics are conveniently divided for study according to the phase affected. Thus compression, combustion, and expansion lines being all high-pressure effects and recorded by high-scale indicator springs Nos. 160 to 300, can be grouped together. Suction and exhaust lines involve pressures relatively low, and as recorded by the high-scale spring, the variations do not show up well, so light springs Nos. 10 to 30 are used. These suction- and exhaust-line changes are not important as affecting the driving of the engine in themselves, but only as enabling the effects observed on the other lines to be interpreted. Thus if a rounded combustion line replace a previously straight one and at the same time exhaust pressures have been found to be higher, then probably neutral dilution of the mixture from excessive back pressure caused the effect.

28. **Form of Card with Different Gas Proportions.**—A card from an ordinary four-cycle Otto engine using carbureted water-

gas of 560 B.T.U. per cu. ft. (high) is shown in Fig. 23. The compression is 73 lbs. absolute and the pressure ratio measured at $^1/_{10}$ stroke is 3.8. Two lines, (*a*) and (*b*), appear near the crest of this diagram; these are the result of the governor action, which is of a hit-or-miss type. Supposing the mixture to contain not quite enough air when engine is running at full load and producing line (*a*), then after a miss the clearance will contain air and supply the deficiency, enabling more gas to burn and raising the line to *b*. Several lines, then, on one card, when the engine has a fixed point of ignition and constant speed, indicate a change in the proportions of the working mixture, which contains air, gas, and hot burnt gas. Every combination of these three gases has a different rate of propagation; more neutral or excess of either air or gas will make the rate of propagation slower than with no neutral. The fastest mixture would be oxygen with some pure gas without inactive material. Using air instead of oxygen adds a large amount of nitrogen and results in a slower mixture, and finally using gases with CO_2, O, steam, etc., and mixing with the burnt gases in the clearance will reduce the rate of propagation still more. Addition of inactive gases, then, to the mixture, either as neutral, excess of air over what the gas present needs, or excess of gas over what the oxygen present will support, results in a flatter card than without dilution, when piston speed and point of ignition are constant. A mixture with excess air gave the card (Fig. 24) in the same engine that produced Fig. 23.

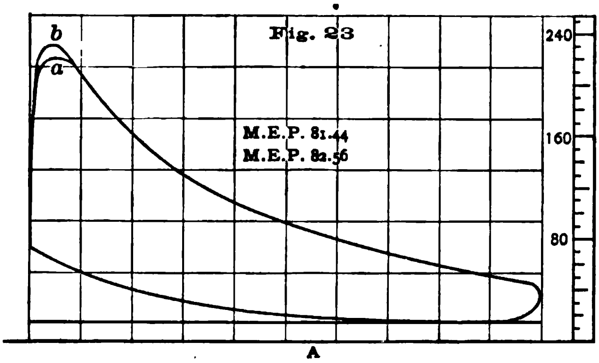
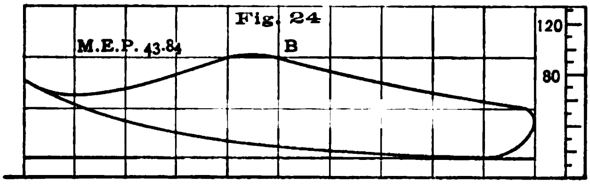
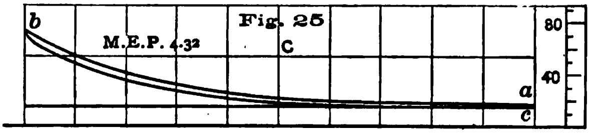


Fig. 23

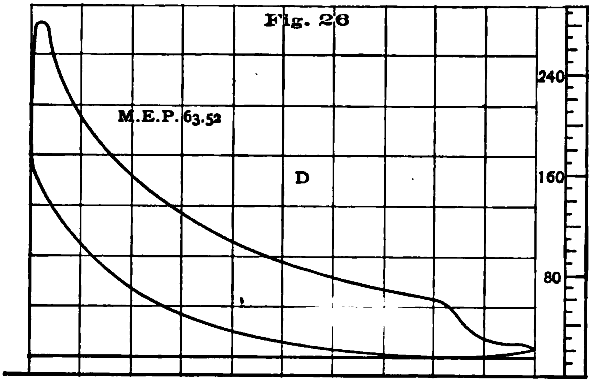
A miss stroke in this engine with hit-or-miss governor results in a card (Fig. 25), the line *ab* indicating compression and *bc* expansion after the miss. The latter is lower because of leaks past piston, valves or ignitor, which are rarely absent in these engines.



When a much-diluted or a naturally weak gas is to be burned in the cylinder some means must be adopted for increasing its rate of propagation and making it more easy to ignite. This is done



with producer and blast waste gases by high compression; a rise of temperature due to compression and the increased density that result produce the effect desired. A card from a large two-cycle



engine on blast-gas is shown in Fig. 26. Here the compression is 170 lbs. absolute on blast-furnace gas of 90 B.T.U. per cu. ft. and the pressure ratio at $^1/_{10}$ stroke is 2. These weak high-compression gases give a higher and thinner card than the richer gases, as may be seen by comparing this card, Fig. 26, with Fig. 23 for

560 B.T.U. gas or with Fig. 27 on natural gas, 1000 B.T.U. This card (Fig. 27) is from a Warren engine with water-cooled piston and valve which permits high compression to be used, 120 lbs. absolute in a cylinder 25″ diameter, at part load. The pressure ratio at $^1/_{16}$ stroke is 3.7, which is lower than it should be because the cylinder has not a full charge, the governor having acted to throttle the supply, and secondly, because the line *a–b*, indicating back pressure, which, however, is easily eliminated, shows there must be confined in the cylinder at the end of exhaust some burnt gases, making the incoming charge already rarefied by the governor somewhat more than usually high in percentage of neutral. Carrying this throttle-governing action still farther, there will come a time when only a very small quantity of air-gas mixture will enter the cylinder and there meet the full clearance charge of burnt gas,

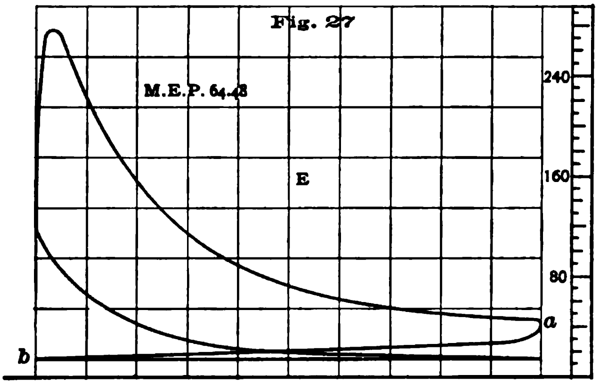


Fig. 27

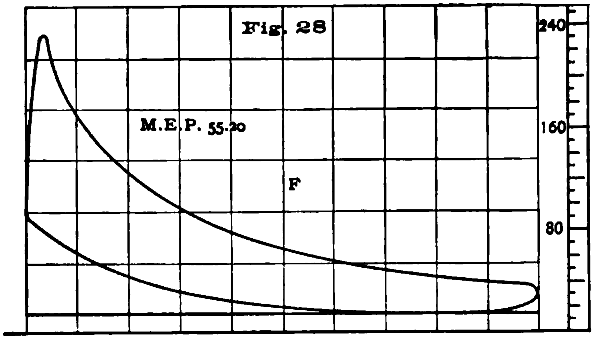
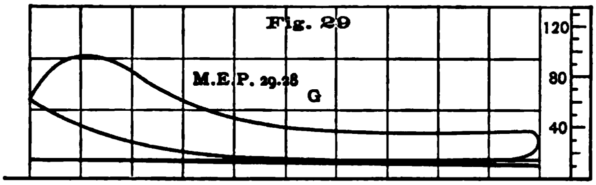


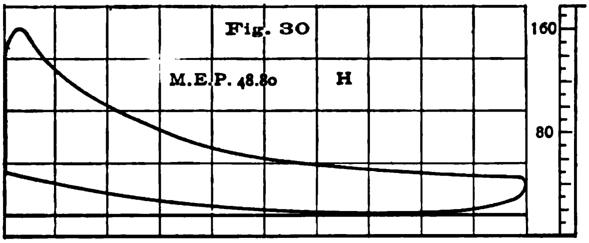
Fig. 28

giving to the resulting charge high in neutral a slow rate of propagation.

A Crossley-engine diagram on coal-gas, 550 B.T.U., with a compression of 90 lbs. absolute and having at $^1/_{10}$ stroke a pressure ratio of 3.4, is shown in Fig. 28, and this is at full load with no throttling. After throttling so that the compression has fallen as a result of initial rarefaction to 55 lbs. absolute, or one-half, the increase in percentage of dilution makes the rate of propagation so slow that with the same speed and ignition the pressure can rise but little above the compression as in Fig. 29. The cut-off governors produce the same effect on the cards.



29. Piston-speed Effects.—The preceding illustrates the fact that the combustion line with constant ignition-point will depend for its form on the rate of propagation of the resulting mixture at constant speed, or more properly, on the relation between rate of propagation and piston speed. The flattening effect due to retarded rate of propagation with constant piston speed will result precisely the same if the rate of propagation be kept constant by regular full charges of mixture, constant in proportion, and the piston speed be at the same time increased. This is shown in Figs. 30, 31, 32. Fig. 30 shows a card on gasolene from a Craig four-cycle engine with a compression of 45 lbs. absolute and a pressure ratio at $^1/_{10}$ stroke



of 3.6. It shows the charge to be slightly throttled; the speed is about 150 R.P.M. By increasing the speed without further adjustment to 500 R.P.M., Fig. 31 results, and at 750 R.P.M. the card

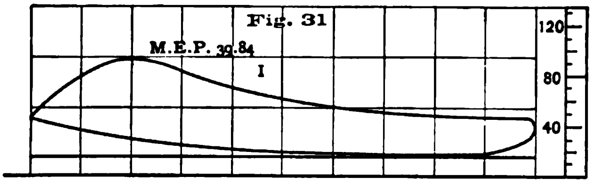


Fig. 31

Fig. 32 results. The fact that the throttling effect also slightly increases at these speeds results in a somewhat rarefied mixture, but this by no means accounts for the rapid fall of the combustion line. It is as if the piston were running away from the flame cap

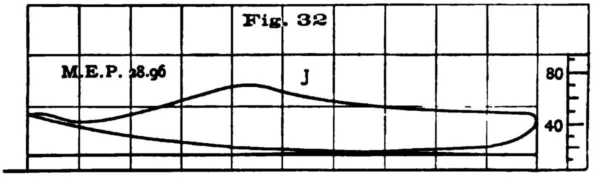


Fig. 32

so rapidly that the combustion could not produce heat quickly enough to keep the pressure up.

30. **Ignition-change Effects with Throttling.**—The third effect that causes serious alterations in the form of the combustion line is the point of ignition. The ignition-point, if kept constant, will permit a flattening of the card with either constant mixture of air, gas, and neutral at increased piston speed or with increase of neutral in the mixture. It is possible, however, to prevent this flattening, at least to some extent, by advancing the ignition-point.

This is illustrated by the cards that follow, which show in nine cards three different points of ignition with three different throttle positions; that is, with three different percentages of dilution, from the rarefied charge being mixed with the same burnt gases on the clearance, and exaggerated somewhat by the back pressure continually present. Of this set, Fig. 33 (*a*) shows a card from an automobile engine with spark properly advanced, the engine burning gasolene, with a compression of 50 lbs. per sq. in. absolute and showing a pressure ratio of 4.0 at $^1/_{10}$ stroke. By keeping throttle and speed the same, that is, by keeping a constant rate of propagation and piston speed, setting the spark just on the center gives card Fig. 33 (*b*). A still later sparking-point gives the card Fig. 33 (*c*). This effect becomes more marked as the percentage of dilution increases by a partly closed throttle, as is shown by the series, Fig. 34 (*a*), (*b*), (*c*), with the same points of ignition and full clearance as in the previous case, but with the incoming charge rarefied.

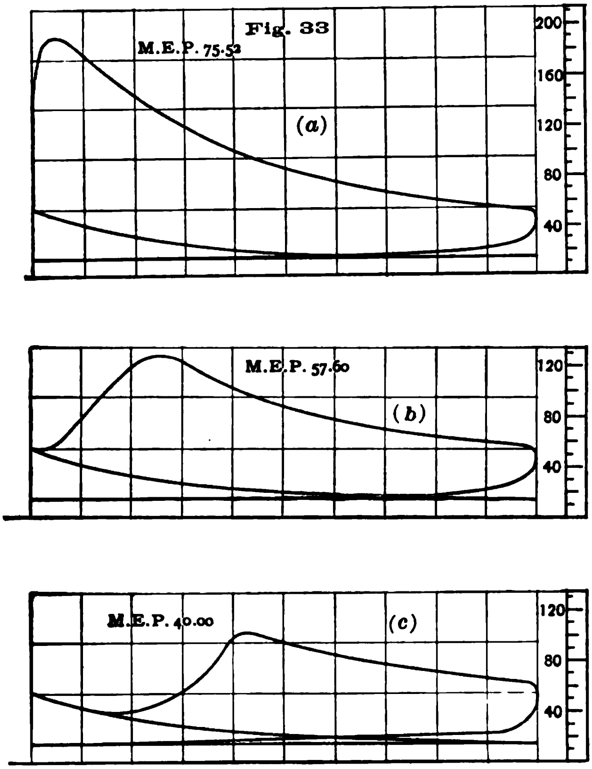


Fig. 33

A still further closing of the throttle gives the series of Fig. 35 (*a*), (*b*), (*c*), which exhibits the effect of still slower rate of propagation with the same three ignition-points. Taking these cards in sets of three, but the other way, cards Figs. 33 (*a*), 34 (*a*), 35 (*a*) show the effect of rarefying the charge by the throttle with proper point of ignition. The set Figs. 33 (*b*), 34 (*b*), 35 (*b*) and Figs. 33 (*c*), 34 (*c*), 35 (*c*) show the same with successively later ignitions. Any of the cards with the flat tops could be raised a little by more advanced ignition, but not entirely brought up to the best and most economical form like Fig. 33 (*a*). This would require not only an advanced spark but also increased compression, whereas they suffer from decreased compression. This spark-throttle variation producing, with but little motion of the links, the widely varying cards shown, furinshes the best means of quickly controlling an engine and is used in automobiles and boats. It must be remembered that this method decreases the economy, first, because energy is lost by drawing the charge through the throttled valve, and secondly, late ignition results in but little work with a full charge. The difference in Figs. 33 (*a*) and 33 (*c*) show the loss of economy due to a late spark. The other loss, due to suction throttling, will show in an apparent decrease in mechanical efficiency, as has been pointed out. It should be clear, then, that the form of the combustion and expansion lines depends on the relation between rate of propagation and piston speed in the first place and between rate of propagation and point

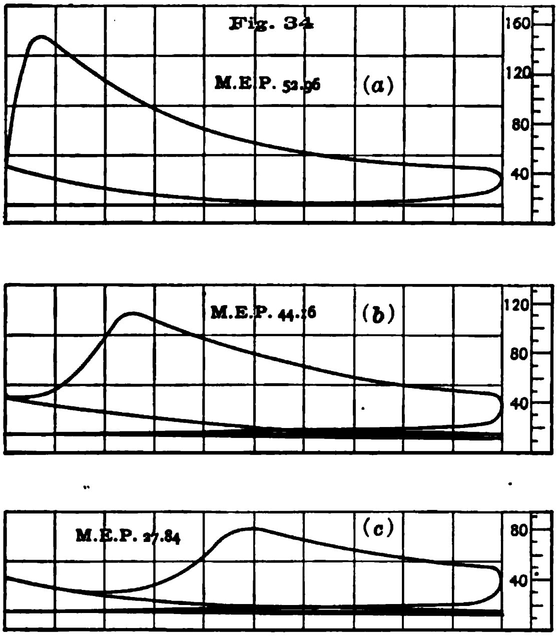


Fig. 34

of ignition in the second. The rate of propagation depends primarily on the percentage of inactive gases in the mixture, and secondly, on the degree to which it is compressed.

The height to which a proper explosion line should rise depends on the B.T.U. per cubic foot of mixture, considering as mixture air, gas, and all neutral present, as already explained. To illustrate the change in form of card with load on a cut-off control

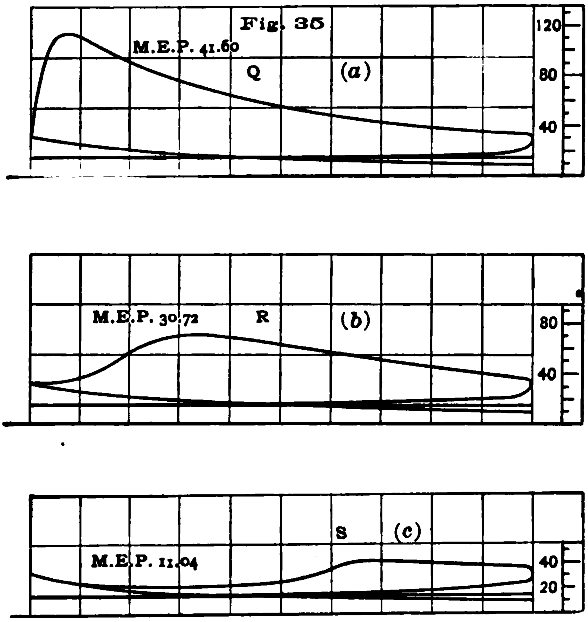
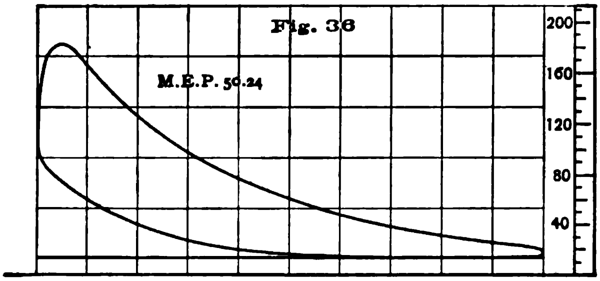


Fig. 35

of mixture, the pair of cards, Fig. 36, from a Sargent engine are presented. It is to be noted that the suction line after cut-off at *A* falls below *B* and the subsequent compression line rises along the same line, involving no loss. However with a constant clearance the percentage of dilution increases, because to constant quantities of burnt gases there is added each time a decreasing amount of mixture. To complete the list of examples for different fuels, Figs. 37, 38, 39 are presented. The first is from the Nacozari-Crossley engines on Loomis-Pettibone producer-gas. The second and third cards, Figs. 38 and 39, are respectively a kerosene card and a gasolene card, such as result when the vapor is

Fig. 36



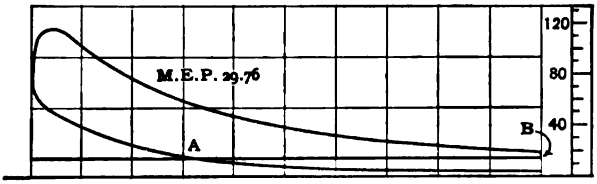


Fig. 37

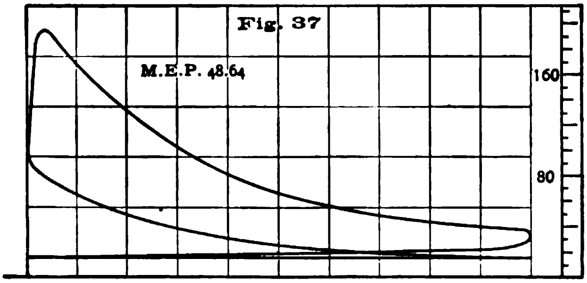
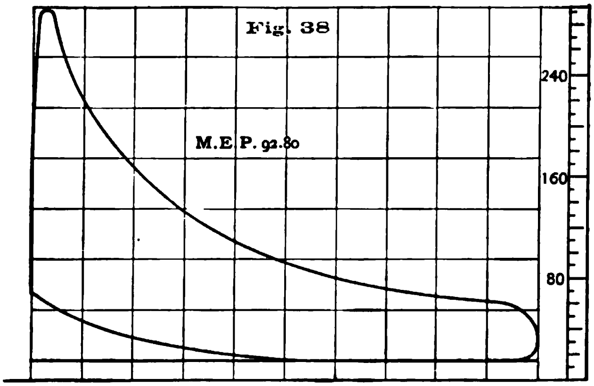
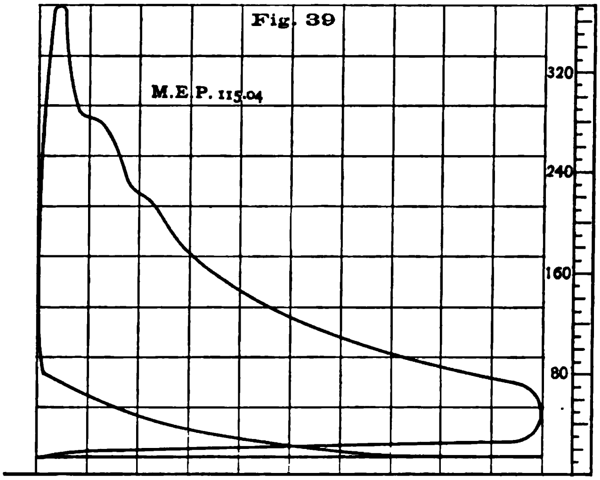


Fig. 38



prepared outside the cylinder without vacuum carburetors and when the charge is a full one. The low-compression and high-pressure ratios are the characteristics of cards on these rich fuels.

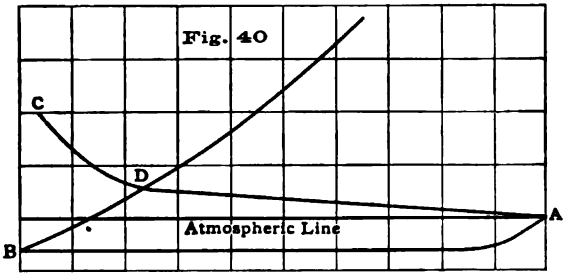


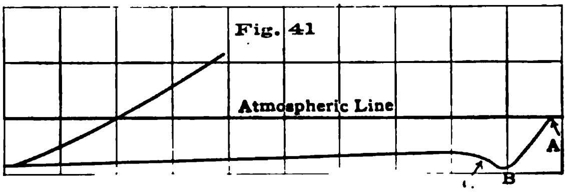
Carburetor or hot-bulb vapor generators would seriously lower the pressure ratios here shown.

31. Pumping Diagrams or Low-spring Cards.—These cards so far presented are on such a large scale of pressure as to make it difficult to read back pressures even as high as 5 lbs. per square inch, as well as the suction, which may go as low as a third of an atmosphere. The pressure changes during these parts of the stroke are comparatively insignificant in producing work-resisting forces in the engine at full loads, but are extremely important in explaining the rest of the diagram. A fall in suction of half an atmosphere will produce a card of flattened form and of an area possibly one-eighth or less of that resulting from a full charge at atmospheric pressure. Similarly a 10-lb. back pressure will, with ordinary compression, practically stop the engine and lower back pressures produce successively flattening cards. The proper or improper setting of the valves can also be read from these low-pressure diagrams of suction and exhaust, while it is practically impossible to tell much beyond ignition setting and exhaust opening period from the regular card. To secure the suction and exhaust lines to a proper scale for interpretation, a low scale spring, 10, 20, or 30 lbs. per square inch, is put in the indicator and a stop inserted to prevent any records above what the indicator can safely handle.

Suction in four-cycle engines takes place in the same cylinder that compresses and fires the charge, whereas in two-cycle engines either the closed crank case or a separate suction cylinder is used. When thus prepared, in crank case or separate suction cylinder, the charge is introduced into the power cylinder during about $^1/_{10}$ of its stroke, at the outer end, by permitting the charge, previously compressed to 5 or 10 lbs. in the suction cylinder, to enter the power cylinder for this period, while its exhaust ports remain uncovered by the piston and the pressure therein remains 1 atmosphere approximately. Thus there is in two-cycle engines no exhaust stroke, but merely an exhaust period of about $^1/_{10}$ stroke, corresponding nearly to the period of charge transfer from suction to power cylinder.

32. Suction- and Exhaust-line Changes.—If the suction-valve be mechanically opened, its lift, when cam-operated, will be somewhat proportional to the piston speed, so that the suction line will be nearly horizontal, as in Fig. 40, starting just beyond *A*, the point where exhaust ended at atmosphere, and extending to *B*, the point where compression begins. Too late an opening of the suction will show if very late, by a quickly falling line, *AB* (Fig. 41) rising to *C*, the mean effective suction line, when valve has opened normally.

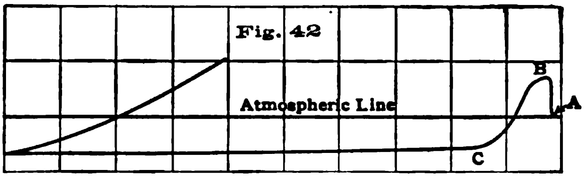




Too early an opening of the suction affects not the suction line but the exhaust, as in this case exhaust gases will enter the mixing-chamber.

In multiple-cylinder engines, however, in which exhausts are coupled together, too early an opening of exhaust of one cylinder may give a line like *ABC* at the end of the exhaust of another (Fig. 42). This is not properly part of the suction line, but a blowing back of an early exhaust from another cylinder.

Fig. 42



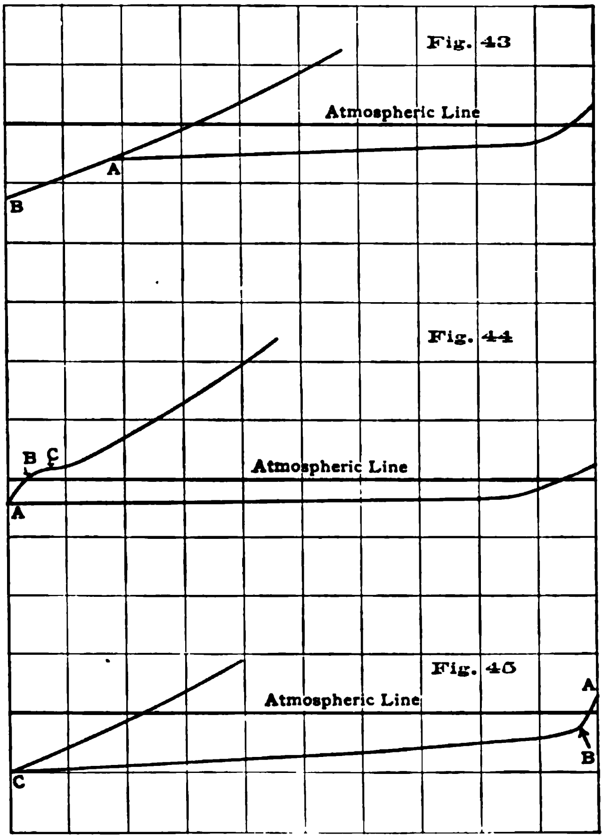
If the valve passages be too small, or the charge be intentionally throttled for governing, the suction line will of course fall lower. Too early a closure of the suction-valve will be shown by a drop *AB* (Fig. 43), due to the expansion of the confined charge and of course coinciding with the first part of the compression line.

Too late a closure of the suction-valve will result in the partial rejection of the charge back into the mixing-chamber at a pressure slightly above atmosphere *ABC* until compression begins (Fig. 44) at *C* after closure.

With automatic suction-valves a doubly rapid falling of the suction line *ABC* (Fig. 45) results, especially if the spring be too stiff or passage too small for the speed. The part *AB* indicates the spring resistance or the pressure below atmosphere necessary to open the valve. The part *BC* indicates too small a valve opening or rather too high a gas velocity.

The form of governor known as the cut-off operates to reduce the charge in the cylinder by sharply causing suction to cease and substituting for the latter part of the suction stroke an expansion of the charge already received, giving suction lines for varying load like Figs. 46, 47, and 48, *B* indicating the point of cut-off.

In two-cycle engines this suction stroke is completed in either a separate cylinder called a pump or in a closed crank case of the engine, and the periods of opening are more or less definitely fixed. With a spring-closed check or mechanically operated valve the suction line is the same in form as for the four-cycle cylinder, except for the shortening that results from the uncovering of the cylinder port by the piston. However, in engines of the class that have two such ports, one from the crank case to the air and another from cylinder to air, with a third connecting crank case and cylinder and all covered or uncovered by the piston, the suction line is a short-pressure rise *BC* (Fig. 49) at the end of the line *AB*, indicating the expansion of the confined crank-case charge. At point *B* the atmospheric port opens and stays open while the piston moves to *C* and back to *D*, in line with *B*, at which point crank-case compression begins.



When partial charges are introduced, the compression line will not begin to rise above atmosphere until after the beginning of the stroke and the point where rise begins gives a measure of the extent

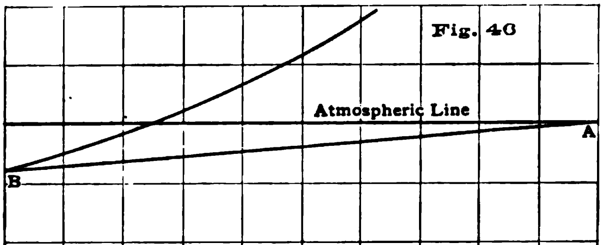


Fig. 46

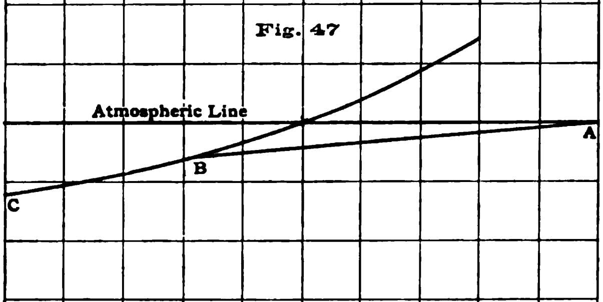


Fig. 47

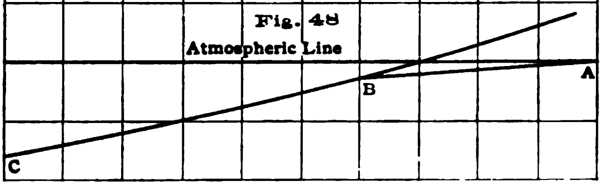


Fig. 48

of the charge. Should the rise begin at half stroke, a half charge has been lost, as in Fig. 47. In this way throttling losses are most easily measured.

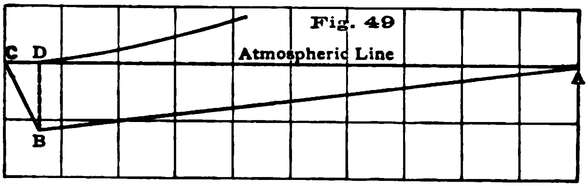


Fig. 49

Exhaust lines will indicate, among other things, two which are of great importance. First, too small an exhaust-valve opening, shown by the exhaust line being continuously above the atmosphere. Secondly, should the exhaust-valve open too late, the expansion line on the high spring card will fail to fall to atmosphere, while the low spring card shows a slowly dropping line to the point where suction begins—line *CD*, Fig. 40.

The forces acting on the piston of an ordinary four-cycle engine are different for every point of the stroke for two successive revolutions. To exhibit these changes, it is necessary to plot these pressures upward from a base line representing for the four-stroke cycle four times the piston path and for a two-stroke cycle twice the piston path. The pressures forming the ordinates are the algebraic sum of the pressures on the two sides of the piston and are usually plotted with reference to the zero or base line, positive, or upward, when they act to assist the motion, and negative, or downward, when resisting the motion of the piston.

33. **Gas Forces on Piston from Indicator Card.**—In single-acting four-cycle engines, on one side of the piston there acts the atmospheric pressure, while on the other side pressures fall below and rise above this. Two-cycle engines in which the suction takes place on one side of the piston or in the crank case, the pressures on both sides of the piston differ continually from one atmosphere, but the pump pressures are usually negligible. Where in a two-cycle engine the suction and precompression take place in one cylinder and the cyclic compression in another, the engine supplied is usually double-acting and the pressure on both sides of both pistons is continually varying. Tandem cylinders should be treated separately and results afterward combined. In any engine, the piston force-diagram will differ with the load, but the nature of the variation is best studied by drawing a full-load and light-load diagram, from which the nature of the curves at any intermediate load can be estimated at a glance or plotted if necessary.

34. **To Draw the Probable Gas Force-diagram.**—The cooling of the walls will prevent the compression line following the law

$$PV^{1.41} = C.$$

The nature of the effect is moreover variable, so that no equation with a constant exponent will exactly represent a real compression line. If, however, as in this case, the line is wanted approximately and its terminal point more especially, it can most easily, and as accurately as possible, be found by using a table of cube roots in connection with the equation

$$PV\sqrt[3]{V}=C.$$

The pressure rise and form of combustion line can be fairly well predicted from what precedes, while the expansion line can be found by applying at successive points the appropriate pressure ratio.

This gives the probable card from which the gas force-diagram can be constructed. These gas force-diagrams are given for various conditions under Turning Effort, where they are combined with the inertia diagrams for the reciprocating parts, to find the forces acting along the connecting-rod.

35. Kinetic Forces.—The movement of a mass when its velocity is increasing requires the impressment of a certain force whose magnitude depends on the mass and the rate at which the change of velocity takes place. Similarly the retardation of velocity of a moving mass causes it to impress a force on whatever causes the retardation. Such forces are termed inertia forces.

Let α = acceleration of mass in feet per second;

W = weight of mass $=Mg=M\times 32.2$;

F = force of inertia;

then will $$F=\frac{W\alpha}{32.2}.$$

Any reciprocating mass, such as a piston or valve stem, generates such forces in an engine; so also do rotating masses with variable angular velocity, such as fly-wheels.

Another kinetic force is met when a mass rotates about a center; this force becomes apparent in the tendency of the mass to move outward radially from the center and is called the centrifugal force.

Let r = radius at which the center of gravity of a rotating mass moves in feet;

v = linear velocity of the mass in feet per second on the circular path;

ω = angular velocity in radians per second $=\frac{v}{r}$;

F_c = centrifugal force in pounds;

then

$$F_c=\frac{Mv^2}{r}=\frac{Wv^2}{gr}=\frac{Wv^2}{32.2r} \quad \ldots\ldots \tag{57}$$

?

$$=.00034\ WN^2r. \quad \ldots\ldots\ldots \tag{58}$$

This can be expressed in terms of angular velocity by

$$F_c=M\omega^2r=\frac{W}{32.2}\omega^2r. \quad \ldots\ldots\ldots \tag{59}$$

Example.—A crank-pin weighs 50 lbs. and the crank radius is 9″, what force is exerted on the cranks at 500 R.P.M. by the pin alone?

$$F_c=.00034\times50\times250{,}000\times.75=3187.5 \text{ lbs.}$$

36. Effects of Kinetic Forces in the Engine.—An ordinary engine has reciprocating parts and rotating parts with constant and variable angular and linear velocities; hence the mere movement of the parts, with or without gas pressures acting, will set up in the mechanism a system of forces which have the following effects.

1°. The reciprocating force of piston, wrist-pin, a part of the connecting-rod and also of piston-rod and cross-head, if the engine be double-acting, will affect the driving force transmitted through the connecting-rod to the crank-pin and modify turning effort; or, to put it differently, there will be a difference between the forces acting on the piston, due to gas pressure, and the force transmitted by the wrist-pin to the connecting-rod.

2°. As a consequence of the above the angular velocity of crank- and fly-wheel will be modified by the inertia of these reciprocating parts.

3°. There will be unbalanced forces and moments acting on the frame of the engine, due to these reciprocating parts, causing shaking and rocking, which, however, may be reduced by balance.

4°. The rotating masses such as cranks and crank-arms will introduce similar shaking and rocking forces and moments in the frame, which may be reduced by balance.

5°. On both these kinds of kinetic forces will depend, at least in part, the strength of certain parts.

6°. Besides the parts that have purely reciprocating or purely rotary motion, the connecting-rods have both motions at once. The kinetic forces due to the connecting-rod exert effects the determination of which involves four problems, as stated by Professor Dunkerly:

a. The finding of a solution for the resultant force to give the rod its combined motion.

b. To find the effect of inertia of the rod on the crank-effort diagram, or to find the "equivalent inertia force" at the wrist-pin.

c. To find the displacing forces on the frame.

d. To find the bending forces on the rod due to its own inertia.

To sum up, then, briefly, these kinetic forces affect

A = the turning effort;

B = the balance or lack of balance in shaking the frame;

C = the stresses in the involved parts, on which their dimensions depend.

It may happen that such distribution of parts and velocities as will give best balance will not give best turning effort; in cases like this a compromise must be made. Stationary engines require, first, a constant turning effort, especially when driving dynamos, while balance is of secondary though certainly of considerable importance. Marine and automobile engines must be balanced first and give as good a turning effort as possible after, but it must be remembered that the turning moment of the shaft is also a turning moment of opposite sign for the frame and there will be a tendency to irregularly rock because of irregularity of turning moment irrespective of all questions of pure balance.

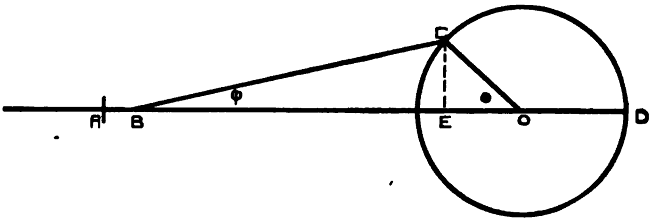


Fig. 50.

37. Inertia Forces of Reciprocating Parts.—*By Approximate Method and with Divided Connecting-rod.*—Let Fig. 50 represent the connecting-rod BC, of length l feet, crank CO, of length r feet, while

$AB = s$ represents the distance the piston has moved from inner dead-center when the crank has moved through angle θ and connecting-rod has moved through angle ϕ.

Put $$\frac{r}{l} = a_1. \qquad (60)$$

Then for uniform angular velocity ω the following is true:

$$\frac{ds}{dt} = \text{velocity of piston along axis;}$$

$$\frac{d^2s}{dt^2} = \text{acceleration of piston along axis;}$$

$$BO = r \cos\theta + l \cos\phi = r + l - s;$$

$$\frac{s}{r} = 1 + \frac{l}{r} - \cos\theta - \frac{l}{r}\cos\phi;$$

$$CE = r \sin\theta = l \sin\phi;$$

hence $$\sin\phi = a_1 \sin\theta;$$

$$\cos\phi = \sqrt{1 - a_1^2 \sin^2\theta}.$$

Developing this by the binomial theorem,

$$\begin{aligned}\cos\phi = 1 & \\ & - \tfrac{1}{2}\, a_1^2 \sin^2\theta \\ & - \tfrac{1}{8}\, a_1^4 \sin^4\theta \\ & - \tfrac{1}{16}\, a_1^6 \sin^6\theta \\ & + \text{etc.} \qquad (61)\end{aligned}$$

Substituting we get

$$\frac{s}{r} = 1 + \frac{1}{a_1} - \cos\theta - \frac{1}{a_1}\{1 - \tfrac{1}{2}a_1^2 \sin^2\theta - \tfrac{1}{8}a_1^4 \sin^4\theta, \text{ etc.}\},$$

or

$$\begin{aligned}\frac{s}{r} = 1 & \\ & - \cos\theta \\ & + \tfrac{1}{2}\, a_1 \sin^2\theta \\ & + \tfrac{1}{8}\, a_1^3 \sin^4\theta \\ & + \tfrac{1}{16}\, a_1^5 \sin^6\theta \\ & + \text{etc.} \qquad (62)\end{aligned}$$

These powers of the sines of θ can be put into sines of multiple angles, for

$$\begin{aligned} &2 \sin^2 \theta = 1 - \cos 2\theta, \\ &2^3 \sin^4 \theta = 3 - 4 \text{ as } 2\theta + \cos 4\theta, \\ &2^5 \sin^6 \theta = 10 - 15 \cos 2\theta + 6 \cos 4\theta - \cos 6\theta, \text{ etc.} \end{aligned}$$

Hence the series above becomes a Fourier series:

$$\begin{aligned} \frac{s}{r} = 1 &+ \tfrac{1}{4}a_1 + \tfrac{3}{64}a_1{}^3 + \tfrac{5}{256}a_1{}^5 + \text{etc.} \\ &-\cos\theta \\ &-\cos 2\theta\,\left(\tfrac{1}{4}a_1 + \tfrac{1}{64}a_1{}^3 + \tfrac{15}{512}a_1{}^5 + \text{etc.}\right) \\ &+\cos 4\theta\,\left(\qquad\quad \tfrac{1}{64}a_1{}^3 + \tfrac{3}{256}a_1{}^5 + \text{etc.}\right) \\ &-\cos 6\theta\,\left(\qquad\qquad\qquad\quad + \tfrac{1}{512}a_1{}^5 + \text{etc.}\right). \quad . \quad . \quad (63) \\ &\quad \text{etc.} \end{aligned}$$

When the angular velocity $\omega = \frac{d\theta}{dt}$ is uniform, a term $C \cos \theta$ represents a harmonic motion completed once in 1 revolution and $C' \cos 2\theta$ represents another of half period, completed twice in one revolution. $C' \cos 3\theta$ represents another of $\frac{1}{3}$ period, completed 3 times in 1 revolution.

Thus the motion of the point is made up of an infinite number of simple harmonic motions of periods which are simple multiples of the engine period. The series for (s) is a convergent one, and for rods greater than 3 cranks in length $\sin \phi$ remains so small that $\sin^2 \phi$ can with fair approximation be neglected in comparison. This gives from equation (61)

$$\cos \phi = 1 - \tfrac{1}{2}a_1{}^2 \sin^2 \theta, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (64)$$

from which

$$\frac{s}{r} = 1 - \cos\theta + \tfrac{1}{2}\, a_1 \sin^2 \theta, \quad . \quad . \quad . \quad . \quad (65)$$

or

$$s = r[(1 - \cos\theta) + \tfrac{1}{2}\, a_1 \sin^2 \theta]. \quad . \quad . \quad . \quad (66)$$

Differentiating and putting linear velocity of crank-pin

$$v = r\frac{d\theta}{dt},$$

$$\frac{ds}{dt} = \text{velocity of piston} = v\,[\sin\theta + \tfrac{1}{2}\,a_1 \sin 2\,\theta];$$

$$\frac{d^2s}{dt^2} = \text{acceleration of piston} = \frac{v^2}{r}(\cos\theta + a_1 \cos 2\,\theta).$$

Put F = inertia force of reciprocating parts in pounds,
M = mass of reciprocating parts,
$W = M \times g$ = weight of reciprocating parts in pounds,
a = area of piston in square inches,
N = R.P.M.,
$\frac{F}{a} = \Big\{$ inertia forces of reciprocating parts in pounds per square inch of piston,
$v = \frac{2\pi N r}{60}$,

we get from the above and article (35)

$$\frac{F}{a} = .00034\frac{W}{a}N^2 r(\cos\theta + a_1 \cos 2\,\theta), \quad . \quad . \quad . \quad (67)$$

at the time when the piston has moved a distance

$$s = r[(1 - \cos\theta) + \tfrac{1}{2}a_1 \sin^2\theta], \quad . \quad . \quad . \quad . \quad . \quad (68)$$

measuring crank-angles always from inner dead-center.

By substituting the values of θ given in Tables XI and XII below the corresponding values of the parenthesis can be found.

TABLE XI.

VALUES OF $(\cos\theta + a_1 \cos 2\,\theta)$.

θ	$\cos\theta + a_1 \cos 2\,\theta$	$\frac{1}{a_1}$ 3.0	$\frac{1}{a_1}$ 3.5	$\frac{1}{a_1}$ 4.0	$\frac{1}{a_1}$ 4.5	$\frac{1}{a_1}$ 5.0	$\frac{1}{a_1}$ 5.5	$\frac{1}{a_1}$ 6.0
0	$(1+a_1)$	1.333	1.286	1.250	1.222	1.200	1.182	1.167
30	$(.866+.5a_1)$	1.033	1.009	.997	.977	.966	.957	.949
45	.707	.707	.707	.707	.707	.707	.707	.707
60	$(.5-5a_1)$	.333	.357	.375	.389	.400	.409	.414
90	$-a_1$	−.333	−.286	−.250	−.222	−.200	−.182	−.167
120	$-(.5+.5a_1)$	−.667	−.643	−.625	−.611	−.700	−.591	−.583
135	−.707	−.707	−.707	−.707	−.707	−.707	−.707	−.707
150	$-(.866-.5a_1)$	−.700	−.723	−.745	−.755	−.766	−.755	−.783
180	$-(1-a_1)$	−.667	−.714	−.750	−.778	−.800	−.818	−.833

TABLE XII.

VALUES OF $\frac{s}{r} = 1 - \cos\theta + \frac{1}{2}a_1 \sin^2\theta$.

θ	$1-\cos\theta+\frac{1}{2}a_1\sin^2\theta$	$\frac{1}{a_1}$ 3	$\frac{1}{a_1}$ 3.5	$\frac{1}{a_1}$ 4.0	$\frac{1}{a_1}$ 4.5	$\frac{1}{a_1}$ 5.0	$\frac{1}{a_1}$ 5.5	$\frac{1}{a_1}$ 6.0
0	0	0	0	0	0	0	0	0
30	$.134+\frac{1}{8}a_1$	.175	.170	.165	.162	.159	.157	.155
45	$.293+\frac{1}{4}a_1$	.376	.364	.356	.349	.343	.339	.335
60	$.50+\frac{3}{8}a_1$	.625	.608	.594	.584	.575	.569	.563
90	$1.0+\frac{1}{2}a_1$	1.167	1.143	1.125	1.111	1.100	1.091	1.083
120	$1.5+\frac{3}{8}a_1$	1.625	1.608	1.594	1.584	1.575	1.569	1.563
135	$1.707+\frac{1}{4}a_1$	1.790	1.778	1.770	1.763	1.757	1.753	1.749
150	$1.866+\frac{1}{8}a_1$	1.908	1.902	1.897	1.894	1.891	1.889	1.887
180	2.0	2.00	2.00	2.00	2.00	2.00	2.00	2.00

38. Plotting Inertia Curves.—From these tables the inertia curve can be plotted if the crank length r be known, if, secondly, the speed M be known, and if, finally, the weight of the reciprocating parts be known per square inch of the piston area $\frac{W}{a}$. It should be noted that inertia is the same in cases where $\frac{W}{a}N^2r$ is the same and a_1 identical.

In single-acting engines the reciprocating parts are simply the piston and wrist-pin, together with whatever water the piston may contain for cooling and the necessary piping connections; in double-acting engines the piston, piston-rod, and cross-head with water and piping, and in tandem engines there must be added another piston with its connecting piston-rod.

The connecting-rod may be considered in approximate calculations as being partly reciprocating and partly rotating. In the diagram of a connecting-rod AB, Fig. 51, let G be the center of gravity distant l_1 from the wrist-pin and l_2 from crank-pin. Then the division may be made so that $m_1l_1 = m_2l_2$, where m_1 = partial mass at wrist-pin and m_2 = partial mass at crank-pin; also $m_1 + m_2 = m$. These m_1 and m_2 partial masses are most easily found by weighing the finished rod on two scales with

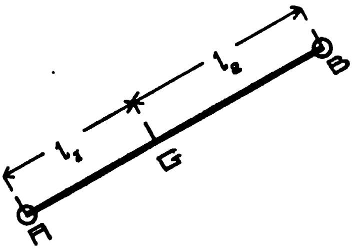


Fig. 51.

each end on a knife edge. Roughly one-half the rod may be assumed as acting at each end, one reciprocating and the other revolving.

This use of the approximation resulting from dividing the rod arbitrarily and dropping the higher terms of the Fourier series for r gives rise to the name of the method as approximate with divided rod.

39. Effect of Rod Length on Inertia of Reciprocating Parts.— The effect of the approximation is best seen in the diagrams, Fig. 52 and Fig. 53, for the data given, the only difference being in the ratio $\frac{r}{l}$, which in one case is $\frac{1}{3}$ and in the other $\frac{1}{6}$. The low value $\frac{1}{3}$ gives an incorrect curve, as it results in inertia pressures near the out dead-center greater than on that center. For ratios greater than $\frac{l}{r} = 5$ the formula is very good, being almost as good as the absolutely correct form.

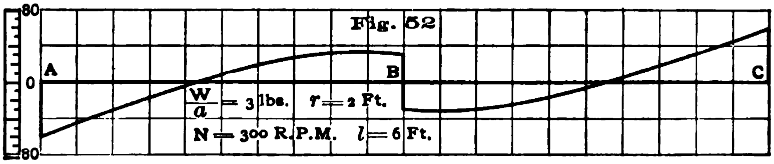


Fig. 52

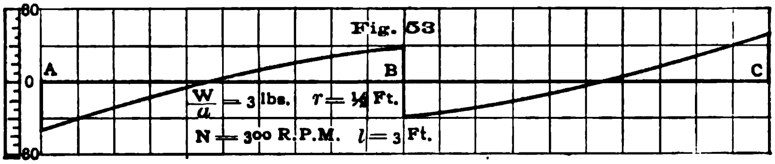


Fig. 53

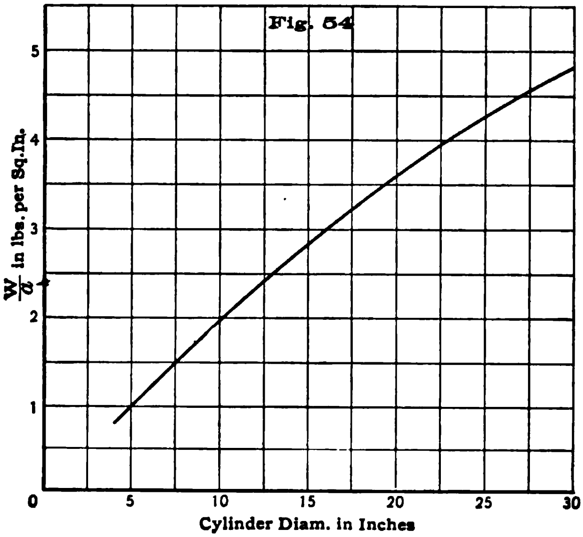
In single-acting gas-engines the weight of reciprocating parts per square inch of piston will vary with the size of the cylinder almost directly. The following curve (Fig. 54) represents average practice well enough to permit preliminary calculations to be made for engines yet unbuilt.

40. Weight of Reciprocating Parts in Gas-engines. — The quantity $\frac{W}{A}$ varying in the average engine from about .6 in the smallest 3½-in. cylinders of automobile engines to about 5.0 in the large 30-in. cylinders of slow-speed stationary engines. The speeds of these engines range normally also from 900 R.P.M. in the smallest to 90 R.P.M. in the largest.

Hence the inertia effects for single-acting pistons will bear the ratio for these two ranges

$$\frac{\text{Inertia of small}}{\text{Inertia of large}}=\frac{C\times .6\times(900)^2}{C\times 5.0\times(90)^2}=\frac{.6\times 10^2}{5}=\frac{60}{5}=12;$$

that is, the inertia forces will be in general about twelve times as great in the small high-speed engine as in the large slow-speed engine.



Double-acting engines with water-jacketed pistons may have reciprocating weights as high as 15 lbs. per square inch of piston area for the large sizes, while tandem cylinders have still heavier reciprocating weights, possibly 25 lbs. per square inch of piston area. Comparing the inertia of the small high-speed engine with the large tandem engine for the speeds 900 and 90,

$$\frac{\text{Inertia of small single-acting}}{\text{Inertia of large tandem}}=\frac{.6\times(900)^2}{20\times(90)^2}=3;$$

so that though the inertia of large stationary engines will be high at times, the maximum is only about a third the maximum for high-speed automobile engines.

It must be understood that these figures for weights of parts will not necessarily fit any one engine, but are rather approximations based on many observations but representing no one case and are useful only in forecasting effects in new engines. The weights for double-acting and tandem cylinders are subject to more variation than for single-acting cylinders.

41. Example in Plotting Inertia on Piston Base.—The method of procedure in the finding of the inertia curve is best illustrated by an example:

Given $\frac{W}{a}=.6$ pounds;
$r=.5$ feet;
$l=2.5$ feet;
$\frac{r}{l}=\frac{1}{5}$;
$N=900$ R.P.M.

From the table of piston positions find s for each 15° of crank angle and set s off opposite θ in a table. Do the same for $\frac{F}{a}$ for the same crank angle θ. This will give the results following.

OUTSTROKE.

θ	s	$\frac{F}{a}$	θ	s	$\frac{F}{a}$
0°	0	99.14	105°	.676	−35.69
15°	.020	94.12	120°	.788	−49.57
30°	.080	79.78	135°	.879	−58.40
45°	.171	58.40	150°	.945	−63.26
60°	.288	33.03	165°	.986	−65.61
75°	.417	7.07	180°	1.000	−66.10
90°	.550	−18.56			

Neglect the ~~crank angles~~ and plot $\frac{F}{a}$ as ordinates and s as abscissæ. This gives the inertia pressures for the outstroke. The return stroke is plotted by reversing the curve, as in Fig. 55. That part of the curve above the zero line represents the energy stored in the reciprocating parts and that below the energy delivered on retardation. These two areas are equal.

It must be remembered that the curves as drawn represent forces necessary to ***produce*** the ***motion*** of the ***piston***. When, however, the ***piston*** is ***driven*** by any means, these forces become resisting and assisting forces, giving the diagram, Fig. 56, as the curve of forces acting on the piston, to be combined with gas-pressure curves to get the effective driving forces at the piston. The curve, Fig. 56, has the same abscissæ as Fig. 55, but ordinates have their signs changed.

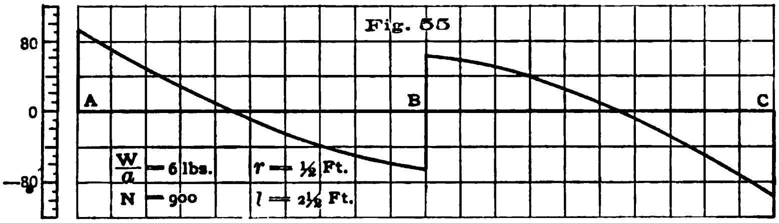


Fig. 55

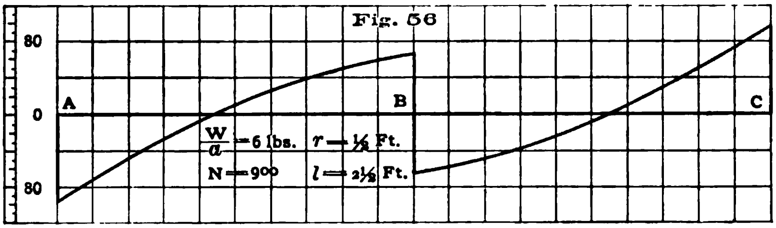


Fig. 56

42. Characteristic Inertia Curves.—To facilitate the work of the designer, a series of inertia curves are worked out for the usual range of conditions in gas-engine work and the results are also plotted. Whenever the product $\frac{W}{a}N^2r$ is found for a given case, the curve can be read off very nearly by comparing with Table XIII here given for different values of $\frac{W}{a}N^2r$.

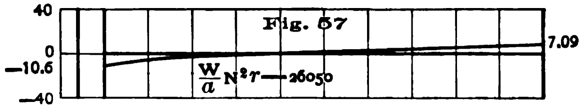
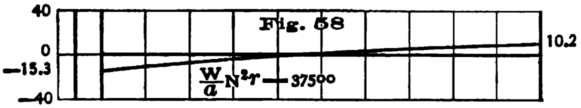
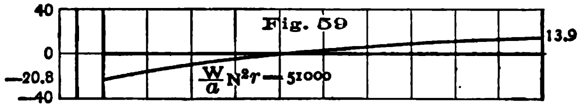


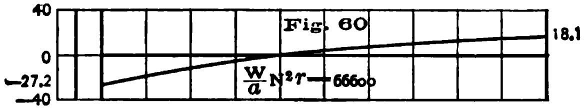
Fig. 57

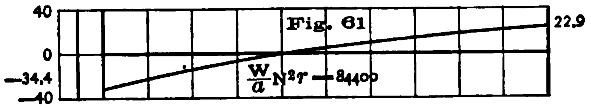
TABLE XIII.

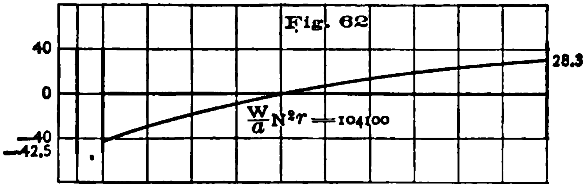
Fig.	......	......	......	57	58	59	60
$\frac{W}{a}$	N	N	N	N	N	N	N
1.0	100	150	200	250	300	350	400
1.5	82	123	163	204	245	286	327
2.0	71	106	142	177	212	248	283
2.5	63	95	127	158	190	221	253
3.0	58	87	116	144	173	202	231
$\frac{W}{a}N^2$	10000	22500	40000	62500	90000	122500	160000
$\frac{W}{a}N^2r$	4165	9370	16680	26050	37500	51000	66600
Initial inertia pressure, lbs. per sq. in.	1.70	3.82	6.81	10.63	15.30	20.81	27.17
Terminal inertia pressure, lbs. per sq. in.	1.13	2.55	4.54	7.09	10.20	13.87	18.12











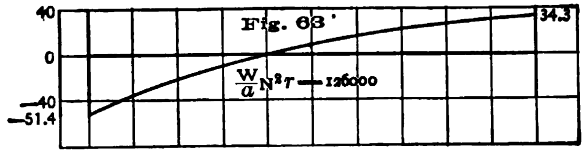


TABLE XIII—(*Continued*).

Fig.	61	62	63	64	65	66	67
$\frac{W}{a}$	*N*	*N*	*N*	*N*	*N*	*N*	*N*
1.0	450	500	550	600	650	700	750
1.5	368	409	450	491	531	572	613
2.0	318	354	389	424	460	492	530
2.5	284	325	357	388	412	443	475
3.0	260	284	318	347	375	404	433
$\frac{W}{a}N^2$	202500	250000	302500	360000	422500	490000	562500
$\frac{W}{a}N^2r$	84400	104100	126000	150000	176000	204200	234200
Initial inertia pressure, lbs. per sq. in.	34.44	42.47	51.41	61.20	71.81	83.31	95.55
Terminal inertia pressure, lbs. per sq. in.	22.96	28.32	34.27	40.80	47.87	55.54	63.70

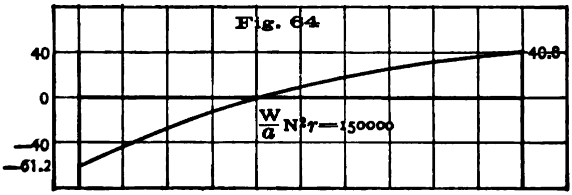


Fig. 64

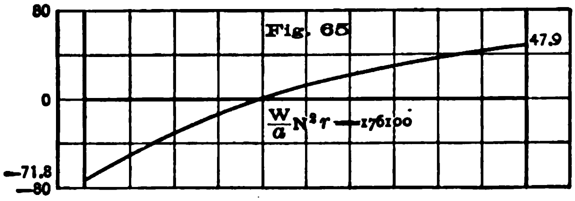


Fig. 65

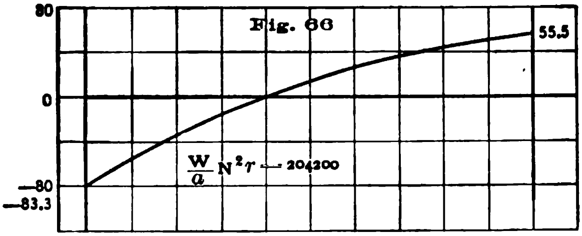


Fig. 66

TABLE XIII—(*Continued*).

Fig.	68	69	70	71	72	73	74
$\frac{W}{a}$	N	N	N	N	N	N	N
1.0	800	850	900	950	1000	1050	1100
1.5	654	694	735	776	816	857	898
2.0	565	601	636	671	707	742	778
2.5	506	538	569	601	633	664	696
3.0	462	490	520	549	578	607	635
$\frac{W}{a}N^2$	640000	722500	870000	902500	1000000	1102000	1210000
$\frac{W}{a}N^2r$	266500	301000	337500	376000	416500	459000	504000
Initial inertia pressure, lbs. per sq. in.	108.73	122.81	137.70	153.41	169.93	187.27	205.63
Terminal inertia pressure, lbs. per sq. in.	72.49	81.87	91.80	102.27	113.29	124.49	137.09

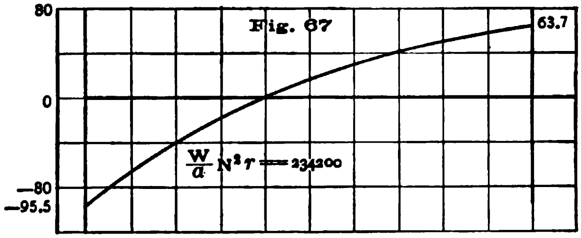


Fig. 67

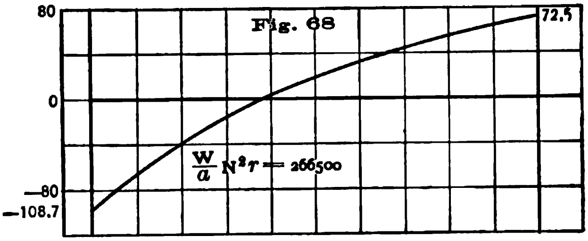


Fig. 68

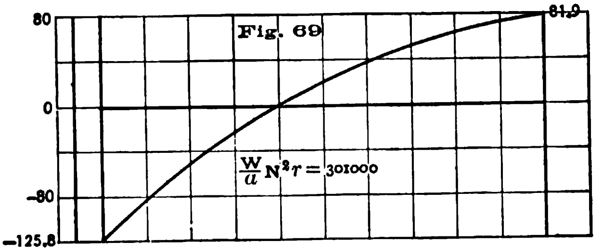


Fig. 69

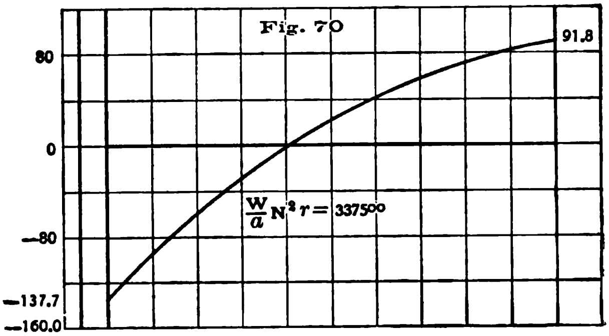


Fig. 70

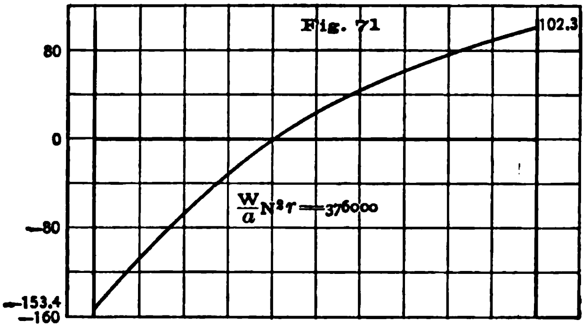


Fig. 71

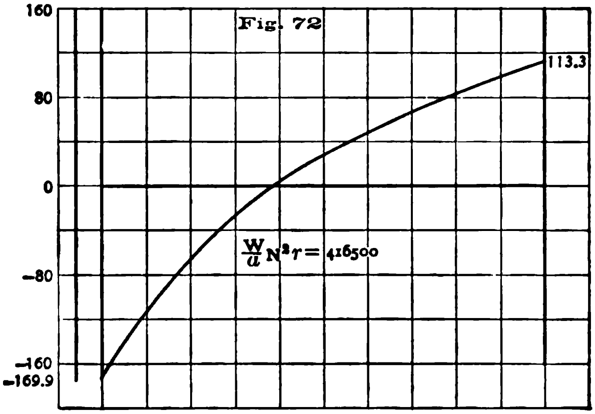


Fig. 72

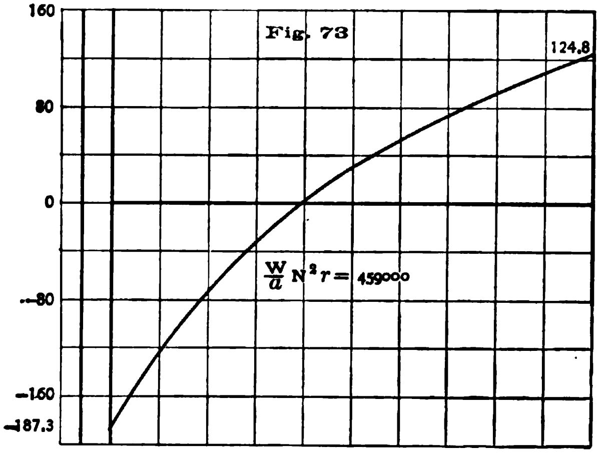


Fig. 73

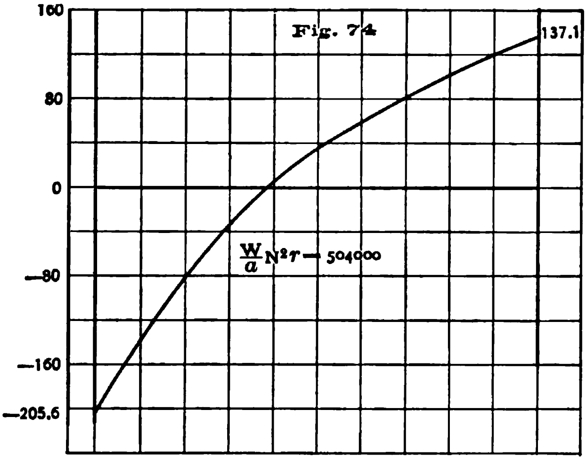


Fig. 74

43. **Effective Driving Forces at Piston.**—*Corrected for Inertia by Approximate Method and Involving Division of the Rod.*—The force acting on the wrist-pin and transmitted thereby to the rod is measured by the difference between the forces acting on the piston by gas pressure and the inertia force.

This difference is the effective driving pressure and the diagrams representing it are effective driving-pressure diagrams.

Let P_e = effective pressure;

P = gas pressure on piston pounds per square inch;

$\frac{F}{a}$ = inertia in pounds per square inch of piston area;

then

$$P_e = P - \frac{F}{a}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (69)$$

The inertia resists gas pressure during acceleration of reciprocating parts or at beginning of each stroke and assists at ends. The effective-pressure diagram is obtained simply by combining ordinates of the two curves. Many examples of this combination are given later, in the process of determining crank-effort diagrams.

44. **Kinetic Forces and Balance.**—Reduction of shaking and rocking forces on the frame by balancing calls for a determination of (*a*) forces due to rotating masses; (*b*) forces due to reciprocating masses.

All kinetic forces and couples not balanced will cause the frame to shake where

(*a*) they are constant in amount and variable in direction;
(*b*) they are variable in amount and constant in direction;
(*c*) they are variable both in amount and direction.

The extent of the shaking forces on the frame is the resultant of all the kinetic forces of the mechanism, and the tendency to rock is the resultant of all kinetic and turning couples. This shaking force is reduced by combining the essential moving parts in different ways, or by adding other masses called balancing masses to reduce the resultant or to change its direction into some less harmful direction, or both. For example, in locomotives where it is impossible to make the resultant zero, the balancing masses are so added as to leave the resultant vertical and so not affect the draw-bar pull, but leaving, however, a pound on the rails. In stationary engines the same is done, the resultant being made vertical by extra weights, to be taken up by the foundation and holding-down bolts. In automobiles the same is true, as the heavy springs can absorb most of the vertical resultant forces; or, better, wherever possible, by proper distribution of masses rather than by the use of counterweights the resultant is made zero.

45. **Balance of Kinetic Forces Due to Revolving Parts.**—The principal revolving masses needing balance in engines are cranks and crank-pins.

In approximate analysis the connecting-rod is divided and part considered as being at the crank-pin as already noted.

Any crank-pin and part of rod weight can be balanced by adding a balance weight, as in Fig. 75, at an appropriate distance.

Each balance weight of W pounds is placed so that the radius R, of rotation for its center of gravity, has the following relation to other quantities.

$$.00034WN^2R = .00034N^2\left[\begin{array}{l}\left(\begin{array}{l}\text{(weight one crank)}\times\text{(distance of}\\ \quad\text{center of gravity from center)}\end{array}\right)\\ +\tfrac{1}{2}\text{(equivalent part of connecting-rod}\\ \quad\times\text{crank radius)}\\ +\tfrac{1}{2}\text{(crank-pin weight}\times\text{crank radius).}\end{array}\right]$$

Let r = crank radius;

r_c = radius of center of gravity of crank;

W_{cp} = weight crank-pin;

W_c = weight crank;

W_r = weight of proper part of rod;

then

$$WR = W_c r_c + W_{cp} r + W_r r. \quad . \quad . \quad . \quad . \quad . \quad (70)$$

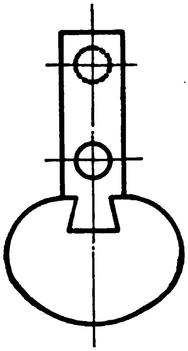


Fig. 75.

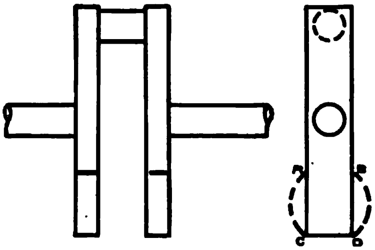


Fig. 76.

Usually R is chosen equal to r for convenience and W found and properly distributed.

The weight W_c can be balanced most easily and directly by simply extending the crank across the center as in Fig. 76. If this is done and if $r = R$,

$$W = W_{cp} + W_r. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (71)$$

Putting all dimensions in inches and pounds and calling the thickness of crank t, the area $ABCD$ necessary for the balance-weight, which we will call A', can be found with the assistance of the table of weights of metals.

	Wrought Iron.	Steel.	Cast Iron.
Wt. 1 cu. in. . . . . $=w$ =	.277	.283	.26
Wt. 1 cu. ft. . . . . $=W_m$ =	480.0	489.6	450.0
Vol. in cu. in. per lb. = v =	3.6	3.53	3.84

$$W=A'\times t\times w;$$

$$\therefore\ A'=\frac{W_{cp}+W_r}{t\times w}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (74)$$

This weight must be symmetrically distributed on both sides of the extended crank, and so that its center of gravity is at crank radius from the center.

It is sometimes desirable to balance the crank and pin by a mass in the fly-wheel. If in Fig. 77 the mass B be placed on the fly-wheel so that $W_b r_b = W_a r_a$, and on the opposite side of the crank, then there will exist static balance—that is, the gravitational forces of the two masses will balance—but this is of no consequence since the centrifugal forces form a couple whose moment is

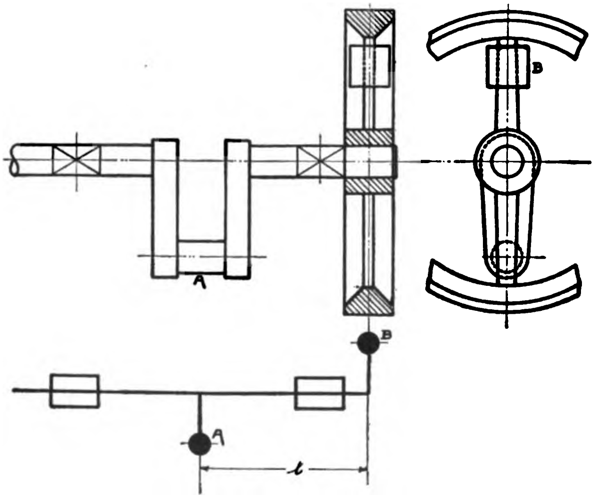


Fig 77.

$$\text{Moment of couple} = 0.00034 W_a r_a N^2 l$$
$$= 0.00034 W_b r_b N^2 l$$

and therefore there is not kinetic balance.

No single rotating mass can balance another kinetically except when applied in the same plane. Application of balance weights in any other planes requires two weights at least, and these may be applied on planes with the original weight between them, as in Fig. 78, where weights are on two fly-wheels, or on planes both on the same side, as in Fig. 79, where weights are added to opposite sides of the spoke of the same wheel. Both these methods may overcome the couple. These masses and their radii can be easily found by the Dalby method.

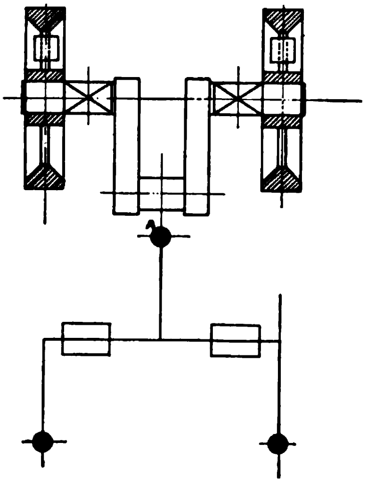


Fig. 78.

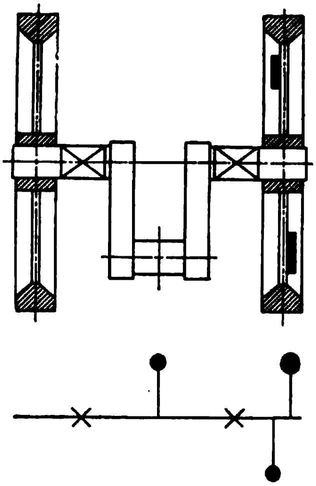


Fig. 79.

46. Balance of Revolving Parts by Masses Placed in Two Planes.—Choose a reference plane in one of the weights to be applied (Fig. 80), for example, plane of revolution for B, thus eliminating one moment from the equation for zero resultant moments. The vector sum of the forces and couple moments must separately be zero.

For no couple about o, the sum of the moments = o, hence

$$\text{Vector sum } (W_a r_a l_a + W_c r_c l_c).00034N^2 = 0. \quad . \quad . \quad (73)$$

The couple whose moment is $0.00034N^2 W_a r_a l_a$ tends to produce rotation about an axis through o, and in the direction of the arrow on the arc through A. The couple whose moment is $0.00034N^2W_cr_cl_c$ should tend to produce rotation about the same axis xx', but in the opposite direction. Moments of couples may be designated by a line called the axis AB (Fig. 81) and which is in length proportional to the moment and drawn on that side of the plane of rotation so as to make the motion when looking toward the plane seem counter-clockwise. Such axes may be combined for resultant moments just as are forces.

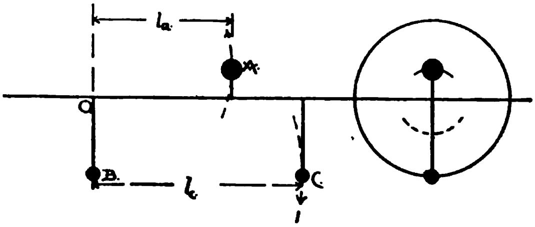


Fig. 80.

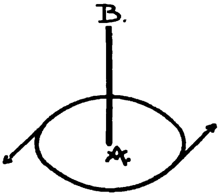


Fig. 81.

In the case just discussed, the two axes would be in the reference plane and at right angles to the radii of the masses, but be opposite in direction. For balance the two axes must be equal in length. This is the statement of equation (73) that the vector sum of the couple moments should equal zero, or

$$W_cr_c = \frac{W_ar_al_a}{l_c}.$$

Fixing on some convenient radius r_c usually just inside the fly-wheel rim, W_c can be found in pounds.

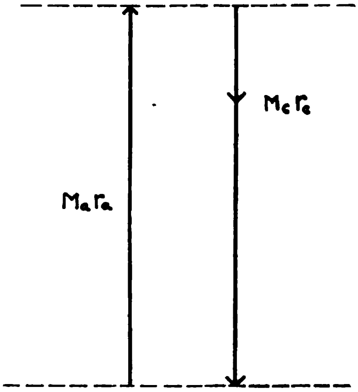


Fig. 82.

There is now no moment about O, but there are the reactions at O equal and opposite to the respective centrifugal forces of WA and WC to be balanced by a force WB.

$$\text{Vector sum } (W_ar_a + W_br_b + W_cr_c)N^2 = O.$$

Hence in Fig. 82 closure gives W_b direct, and the magnitude can be found from the equation of the scalar sum.

47. **Example.**—Suppose a crank and pin with part of connecting-rod to weigh 50 lbs. and two fly-wheels at planes equidistant and each 10″ from crank-pin center; center of gravity of crank to have a radius of 5″ and radii of centers of gravity of masses to be applied, 20″;

then
$$\begin{aligned}
W_a &= 50 \text{ lbs.};\\
W_b &= \quad ;\\
W_c &= \quad ;\\
l_a &= 10'';\\
l_c &= 20'';\\
l_b &= 0;\\
r_a &= 5'';\\
r_b &= 20'';\\
r_c &= 20''.
\end{aligned}$$

Hence
$$W_c = \frac{W_a r_a l_a}{r_c l_c} = \frac{50 \times 5 \times 10}{20 \times 20} = \frac{25}{4} = 6.25 \text{ pounds};$$
$$W_b r_b = W_a r_a - W_c r_c;$$
$$W_b = \frac{(50 \times 5) - (6.25 \times 20)}{20} = \frac{125}{20} = 6.25 \text{ pounds}.$$

Finding balance weights when the fly-wheels are unequally distant, as they usually are in practice, to make room for the half-speed gear and driving-pulley for the following conditions, gives unequal weights:

$$\begin{array}{lll}
W_a = 50 & l_a = 10 & r_a = 5\\
W_b = & l_b = 0 & r_b = 20\\
W_c = & l_c = 24 & r_c = 10
\end{array}$$

From which $W_c = 10.42$ lbs. and $W_b = 7.43$ lbs.

This general method of using two planes of application for the balance weights of revolving masses and determining the location and mass of these weights by taking a reference plane and one balance weight and

making vector sum of centrifugal forces = 0,
" " " " " couples = 0,

as developed by Professor Dalby, is always available, but is not so direct as the method of individual weights for each crank. It becomes possible in a four-crank engine to select the two planes of application as those for the two outer cranks, and when weights and distances are very unequal, as they are in multiple-cylinder marine engines, the system can be balanced in itself by properly adjusting the weights of the two outer cranks and determining the proper angles between the cranks. The great value of the method is not in its application to revolving weights purely, but in the balancing of reciprocating inertia forces of the first period. The unbalanced force in a system of revolving masses is the vector sum of the forces acting or is the closure of the centrifugal-force polygon. The unbalanced centrifugal couple is the vector sum of the centrifugal couples or is the closure of the polygon of couple axes.

47. Balance of Kinetic Forces from Reciprocating Parts with Approximate Formula and Divided Rod by Revolving Parts.—If the reciprocating parts be imagined acting horizontally at the crank-pin, which of course they do by the rigid-rod transmission, and if a revolving mass be placed on the crank opposite, a partial balance will exist. For the horizontal component of centrifugal force (Fig. 83), AB may be made equal at any point to the first-period horizontal inertia forces of the reciprocating forces DC.

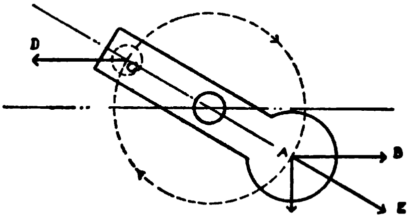


Fig. 83.

But two facts in this connection must be noted:

1°. There is still an unbalanced vertical component of the revolving balance-weight.

2°. If the horizontal components balance at some angles they will not at others, as can be easily shown.

It was found that for approximate analysis the inertia force was given by

$$F = 0.00034WN^2r\ (\cos\theta + a_1 \cos 2\theta).$$

Calling w the rotating balance weight, then its horizontal component at radius ρ is

$$0.00034 w N^2 \rho \cos \theta.$$

If these two horizontal forces are to be equal

$$Wr(\cos \theta + a \cos 2\theta) = w\rho \cos \theta;$$

$$\therefore \; w = \frac{W_r(\cos \theta + a \cos 2\theta)}{\rho \cos \theta}.$$

The changes in w for the different angles is shown by the table.

θ	w
0°	$\frac{Wr}{\rho} \times (1 + a)$
45°	$\frac{Wr}{\rho} \times 1$
90°	$\frac{Wr}{\rho} \times \frac{-a}{0}$
135°	$\frac{Wr}{\rho} \times 1$
180°	$\frac{W\rho}{r} \times (1 - a)$

The fact that at 90° there is no horizontal component of the centrifugal force makes the quotient $\frac{Wr}{\rho} = \infty$ or indeterminate. It is plain that no revolving weight placed counter to the crank-pin can ever balance the reciprocating parts whose inertia is given by this formula.

When more exactly given the inertia-force formula is a Fourier series, and given by

$$\begin{aligned} \frac{F}{A} = c(\cos \theta + c' \cos 2\,\theta \\ + c'' \cos 3\,\theta \\ + c''' \cos 4\,\theta + \text{etc.}). \end{aligned}$$

For the approximation used throughout this work terms above $\cos 2\,\theta$ are dropped, but if retained, as they should be for exact work, the equation above will show how very far from perfect balance is a rotating part when opposed to a reciprocating part on a finite connecting-rod. Dropping all but first-period terms and so neglecting all but first-period forces, it will appear that

$$w = \frac{Wr}{\rho},$$

and this shows that forces of the first period may be balanced by revolving parts, but none of higher period. Sometimes this is sufficient, but often it is not, and when not, other devices must be used.

48. Balancing Reciprocating Parts by Similar Parts.—A perfect balance can be secured by introducing masses with inertia forces of second, third, fourth, etc., period or by directly opposing exactly similarly moving parts. John Macalpine, who devised a reciprocating system shown in Fig. 84, has stated the means for securing the required periodic forces:

1°. The use of revolving weights driven at the required speeds by gear-wheels.

2°. Bob weights in which the ratio of length of connecting-rod to stroke of the weight is properly determined.

3°. Weights made to move in the required manner by other linkages than the simple rod and crank or eccentric.

4°. Weights to which the required movement is given by cams.

None of these systems has ever been used with gas-engines, but it is by no means sure that they may not.

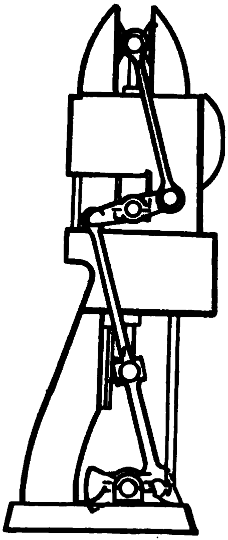


Fig. 84.

They are desirable or necessary only when the service rigidly fixes the arrangement of cylinders, as the screw-propeller drive on ships limits the arrangement to numerous parallel cylinders at various crank angles.

With gas-engines, as used up to the present, a much greater elasticity of arrangement of ordinary parts has been practiced to secure balance, no bob weights or variable piston weights on odd crank angles being used.

A reciprocating mass can be balanced only by another reciprocating mass so placed as to leave no free couples and no free forces.

49. Two Pistons on Opposed Cranks.—Consider a pair of cylinders (Fig. 85) on cranks opposed and side by side. The inertia curve of *A* for the outstroke is given in Fig. 86, and represents the equation $\frac{F}{A}=0.00034\frac{W}{A}N^2r(\cos\theta+a\cos 2\theta)$. At the same time the piston *B* is on its return, giving the curve *B*, Fig. 86. The algebraic sum of these ordinates is not zero; that is, the inertia forces are not equal nor are they opposite. If they acted on one line there would be no couple; however, their lines of action are separated by a distance equal to the distance between their cylinder centers. This will form a couple which has the arm *d* and whose force is given by the difference between the inertia curves.

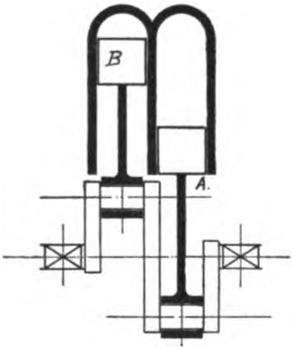


Fig. 85.

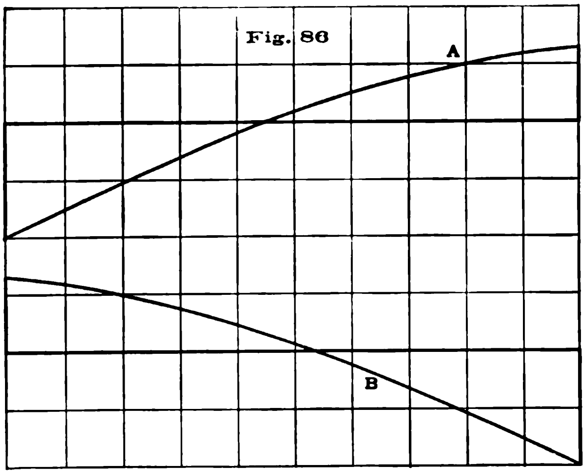


Fig. 86

This moment is a maximum every time a piston starts toward the crank; that is, twice per revolution. As it is zero at some intermediate point, the effect will be a shaking of the frame as well as a rocking. The closer these two cylinders center lines, the less the couple and the less the rocking. Thus in Fig. 87, by placing the cylinders on opposite sides of the crank-shaft, the arm of the couple, instead of being about 1½ cylinder diameters, as it often is in small engines to account for the jacket and walls, has become the distance between connecting-rod ends which is equal to the length of a single crank-pin. By forking one rod, as in Fig. 88, the inertia forces are directly opposed and the arm zero and the couple eliminated, though the resultant forces are not zero.

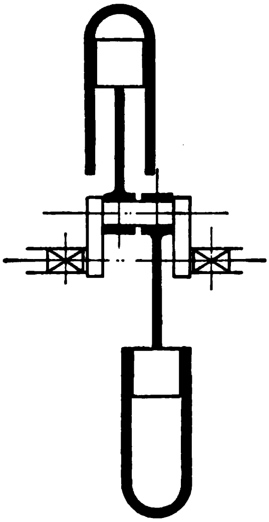


Fig. 87.

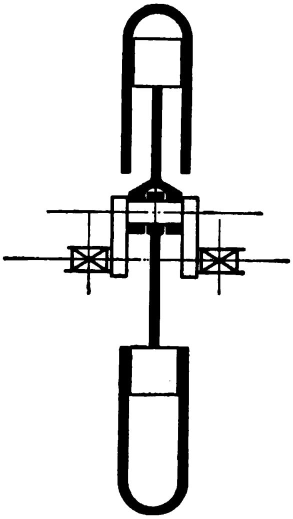


Fig. 88.

50. **Other Arrangements for Two Pistons.**—By using two cylinders whose cranks are not opposed, balance will not result. Take for example a 90° crank angle. Then (Fig. 89) the inertia curve of A will be combined with the inertia curve of B displaced by the distance l, or the path of the piston for a 90° crank movement, leaving forces acting whose amount would be the algebraic sum of the two inertia curves, Fig. 90, and no couple if the cranks were forked as in Fig. 91, but which in case of Fig. 89 leave an unbalanced couple as well. The difference between these cases is not simply the pressure or absence of the couple, as the resultant free forces are not zero.

Two pistons, as in Fig. 92, with three equal cranks give nearly balanced forces and no couple. The differences which exist arise from the different rod lengths, even when masses are equal. The

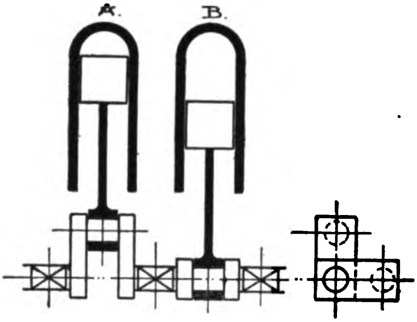


Fig. 89.

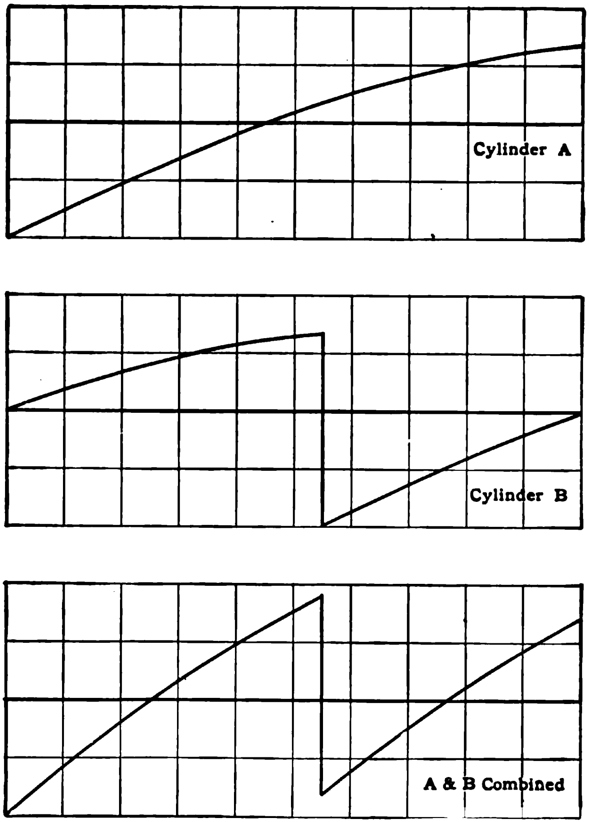


Fig. 90

inertia formula, it will be remembered, involves the rod to crank ratio. Making a_1, the same in both reduces the case to that of Fig. 85. However there is still a disturbing force whose amount is the difference between the inertia curves. In combining, the incenter of one must be combined with the outcenter of the other, as in Fig. 93.

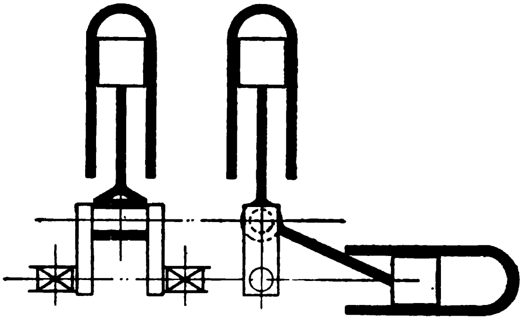


Fig. 91.

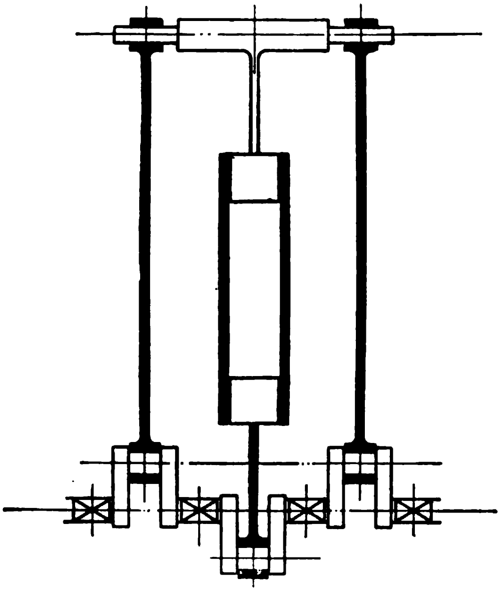


Fig. 92.

The fact that there is a difference at all is due to the fact that the angles of the two rods are not equal for any given crank angle and therefore pistons have not equal acceleration for some crank angles.

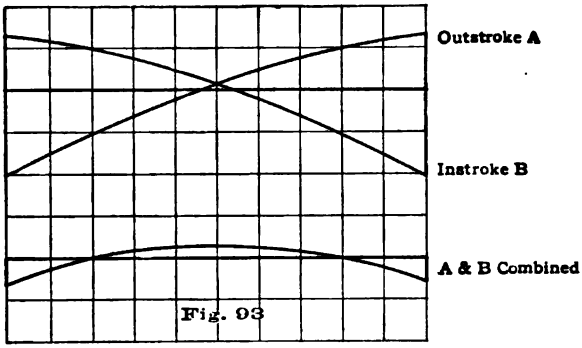


Fig. 93

51. Three-crank Balance.—Three equal cranks when set opposed so that the central crank opposes the two outer ones, and where the reciprocating masses of the center equals twice that at either side, is an arrangement that gives no couple, but leaves unbalanced forces, as in the case of Fig. 92, equal to the combined inertia diagram.

When three cylinders are arranged with axes at 120° and all in same plane with forked connecting-rods, there is the same mass acting at the same radius and same speed at three equal angles.

Call these angles of the crank with respect to each cylinder axis θ, $\theta+120°$, $\theta+240°$. These three inertia forces are then added.

$$\frac{F}{A}=0.00034\frac{W}{A}N^2r\left[\begin{array}{l}\cos\theta+\cos(120°+\theta)+\cos(240°+\theta)\\+(\cos 2\theta+\cos 2(120+\theta)+\cos 2(240°+\theta)\end{array}\right].$$

But

$$aO=OC+Ob.$$

In Fig. 94 it appears that

$$\cos\theta+\cos(120°+\theta)+\cos(240+\theta)=O,$$

and similarly

$$\cos 2\theta+\cos 2(120+\theta)+\cos 2(240+\theta)=O;$$

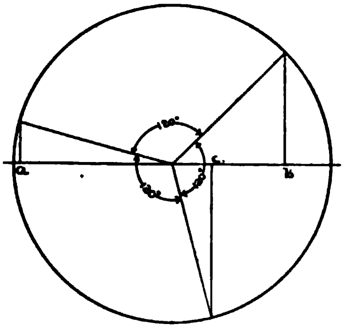


Fig. 94.

Hence for this case there is perfect balance of the purely reciprocating parts, since no free forces and no couples exist.

By placing the three cylinders in line with three cranks, at 120° there is introduced a couple, with the same three forces, however, balanced otherwise in themselves.

52. **Four or More Crank Balance.**—Four cylinders when arranged as shown in Fig. 95 will give no couple, as the couple arising from cylinders *A* and *B* is balanced by that of *C* and *D*, acting to turn the frame in the opposite direction. But there are free forces equal to twice the combined inertia diagrams of *A* and *B*. In Fig. 96 the curve *aa* is the inertia diagram of *A* on outstroke, *bb* that of *B* on instroke, *xx* the combined diagram, and *yy* the same with double ordinates. The forces themselves can be eliminated with the couple by radial cylinders on one crank if all work in one plane.

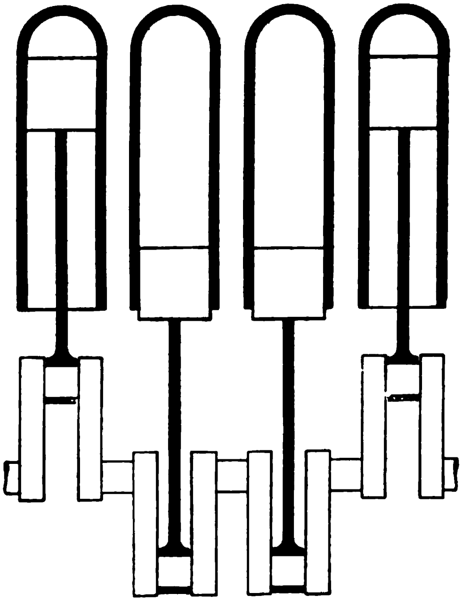


Fig. 95.

More than four cylinders makes balancing of reciprocating parts easier. Five cylinders and equal cranks with a double mass in the center will do, and six cylinders doubling three is also free of forces and couples from purely reciprocating parts, as the couple of that case meets another similar and opposed.

53. **Special Arrangements for Balance.**—Various odd methods of connection have been used for balance, but usually with more

bad points than good ones. Of this class is the double-gear crank discs, Fig. 97. In this it is readily seen that by revolving counterweights the first-period inertia forces can be separately balanced,

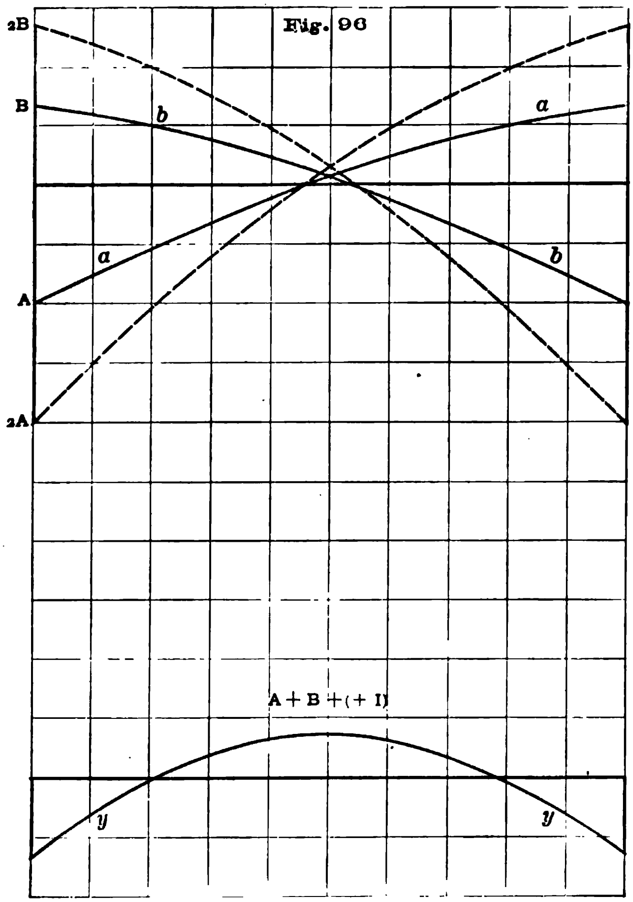


Fig. 96

while the usually unbalanced components perpendicular to cylinder axis are balanced one by the other at their respective bearings by the reversed rotational direction.

Opposition of the pistons, but still dependent on gears, is done as shown in Fig. 98. Oppositely moving pistons with exactly equal rods, cranks, and angles and all in the same plane, hence producing no rotating couple, is secured by the device of Fig. 99—large driving shaft. Equal acceleration is secured by a number of links, all having weight, and two cranks necessarily out of line, so that a slight couple will exist. Each additional moving part calls for more service from the crank, since each has some inertia.

Another and much more common method of securing the oppositely moving pistons is shown in Fig. 92. Here unequal rod angles mean unequal piston acceleration and hence only partial balance. The introduction of side cross-heads to give equal rods would, if weights were kept equal, produce fair balance.

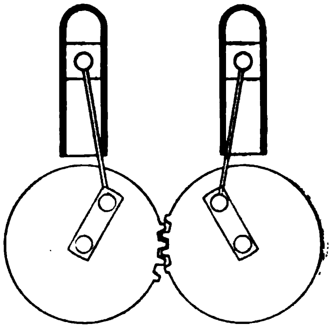


Fig. 97.

54. **Shaking, Rocking, and Vibration of Frames.**—From what precedes it will appear that the kinetic forces of the mechanism which may be balanced wholly, partly, or not at all are of the following classes:

1°. Forces due to the inertia of the reciprocating parts in the direction of the cylinder, which may be balanced by other similar reciprocating parts properly placed, but in no other way.

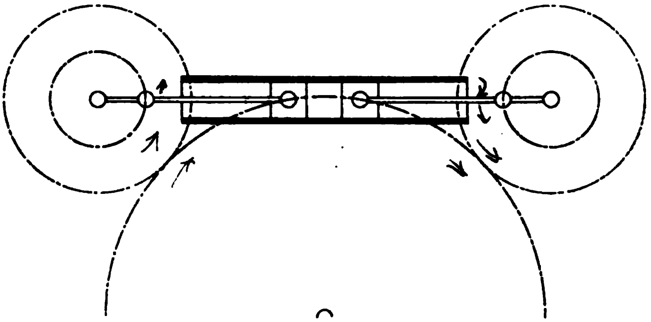


Fig. 98.

2°. Centrifugal forces due to cranks, crank-pins, and other truly rotating parts and which are easily balanced completely.

3°. Centrifugal forces due to the connecting-rod which may be balanced only by a similar and oppositely moving piece.

4°. Centrifugal forces due to counterpoises for reciprocating parts which always have free forces.

5°. Couples due to combination of the preceding forces acting along different lines.

6°. Couples due to the inertia of the fly-wheel and other rotating parts.

7°. Vibrations due to yielding of frame at slides and yielding of supports.

Shaking will result from forces which are:

1°. Variable in amount but constant in direction, such as result from the reciprocation of the piston, etc.

2°. Constant in amount but variable in direction, such as the centrifugal force of a revolving mass.

3°. Variable in amount and direction, such as result from true connecting-rods.

A single force such as results from a pure reciprocation tends to lift and drop the bed when the engine is vertical, and the period of the force will be the period of the shaking that will result. As the true inertia for a finite rod is given by a Fournier series, the lifting will be a force of many periods; or, better, there will be many such lifting and dropping forces occurring at time intervals:

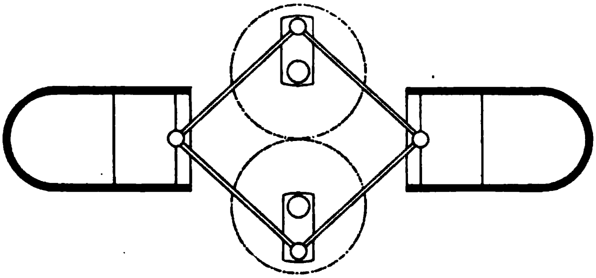


Fig. 99.

a – equal to that of the engine revolution;
b – " " one-half the engine revolution;
c – " " one-third the engine revolution, etc.

Should any part of the frame be flexible it might have a natural period of vibration, just as has a bar, a string, or a spring. Just what is the natural period of vibration of the frame it is impossible to calculate, but by remembering that the first-period forces from the reciprocation are greatest, and that the stiffer the frame, and therefore the shorter its period, it appears that for stiff frames synchronism between the inertia forces and frame periods will occur only with the higher period and so with the smaller inertia forces. It is for this reason that irregular, stiff frames are preferable to simple flexible ones, and frames are designed for stiffness and short period of vibration rather than merely for strength.

When a shaking force acts on a part of a frame held by a bolt at one point, there is at the bolt an equal opposite force, and the two forces acting at a distance tend to turn the frame. Thus a rocking or tilting couple results. Two active forces in the frame itself, regardless of the holding-down counter forces, may produce these tilting or rocking couples.

The above cases exist when there are unbalanced forces, but should all the kinetic forces be balanced, there would still be a rocking couple equal to the turning moment at the shaft. As the shaft is driven in one direction the engine frame is driven in the opposite direction; that is, the frame tends to turn about the shaft. If this turning moment be perfectly uniform, there will be no tendency on the part of the frame to rock, but if the couple be very irregular, such as results, as will be seen later, in a single-cylinder four-cycle gas-engine, the frame will be driven backward with a variable force whose moment equals the turning moment, and will cause a rocking effect, and with a springy support, vibrations of the whole system.

The periodic changes of turning moments compared with the natural period of the frame and supports will determine just how serious will be this kind of vibration, for it is only when in synchronism that these periods result seriously. The periodic change of turning moment will be examined later for various conditions in gas-engines.

55. Shaking Forces and Crank-angles.—The kinetic forces that produce these shaking, rocking effects and resultant vibrations are best examined by dividing them into two systems, one axial or parallel to the cylinder axis, and the other perpendicular to this. These two systems will be called the axial and the transverse.

Inertia forces of reciprocating parts are always axial. Centrifugal forces are to be divided into axial and transverse components; thus cranks, part of connecting-rods, crank-arms, and balance weights will give two components.

Centrifugal force $= 0.00034WN^2r$.
Axial component $= 0.00034WN^2r \cos \theta$;
Transverse component $= 0.00034WN^2r \sin \theta$.

Plotting these forces from two axes at right angles to each other and vertical forces upward or downward through each point or crank-angle will give a curve. In Fig. 100 lay off equal crank-angles giving points *abcdef*, etc., for 0°, 30°, etc., of crank from the axis of cylinder *CD*.

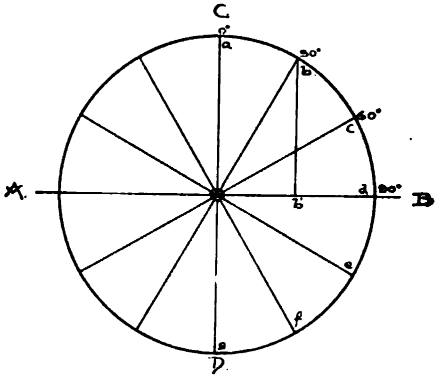


Fig. 100.

Then at *O* lay off *Oa* and *Og* equal to $0.00034MN^2r$.

Verticals *bb'*, *cc'* will then represent

$$bb' = 0.00034WN^2r \cos 30°,$$
$$cc' = 0.00034WN^2r \cos 60°, \text{ etc.};$$

so that the circle represents the curve of centrifugal forces. Axial components at any angle will be the distance from the point to the base *AB*, while transverse components are given by perpendiculars to the base *CD*.

This curve can also be laid off from a base line divided into equal parts, each part representing an equal angle. This will give two curves, one a sine curve and the other a cosine curve, of equal amplitude.

In Fig. 101 divide the base *AB* into twelve equal parts, each representing 30°. Lay off *aa'* equal to $0.00034MN^2r$, corresponding to cos 0°; also *gg'*, corresponding to cos 180°; also *mm'*, corresponding to cos 360°. Then complete the cosine curve passing through *d* at 90° and *j* at 270°. This curve will represent axial force due to the revolving mass at each crank-angle. Similarly draw for the same angles the sine curve, which is the same as the above displaced 90° and having same amplitude. In Fig. 101 the dotted line will then represent transverse forces due to revolving mass.

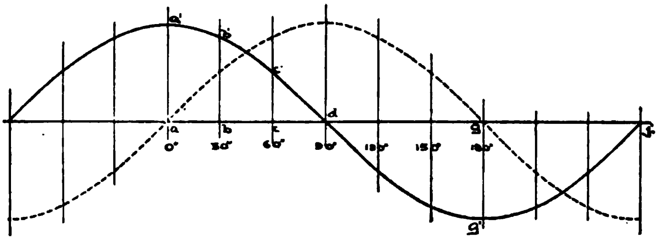


Fig. 101.

56. First- and Second-period Reciprocating Inertia Force and Crank-angles.—In a similar way the axial force, displacing the frame, due to the reciprocating mass, can be laid down. The force is given by a Fourier series of several successively smaller periods, but in the approximate analysis only the first and second have been retained, and this leaves

$$F = 0.00034WN^2r\ (\cos\theta + a_1\cos 2\theta),$$

where $$a_1 = \frac{r}{l}.$$

This can also be plotted directly from the circle, as in Fig. 102, by dividing the work into two parts. The first part, involving the first-period force only, is precisely the same as for revolving masses. Hence on a circle whose radius is $0.00034WN^2r$ the abscissæ Oa, $b'b$, $c'c$, etc., represent the inertia forces of the first period. To get the second-period forces add to each abscissa a fraction $\frac{r}{l}$ of the abscissa of twice the angle.

Thus to bb' for 30° add $\frac{r}{l}\times cc' = bb''$.

When this addition is done algebraically the curve results. Using a straight-line base with equal spacing for equal angles makes the effect of the second-period force clearer. In Fig. 103, $ab = bc = cd$, each of which represents 30° and aa' represents

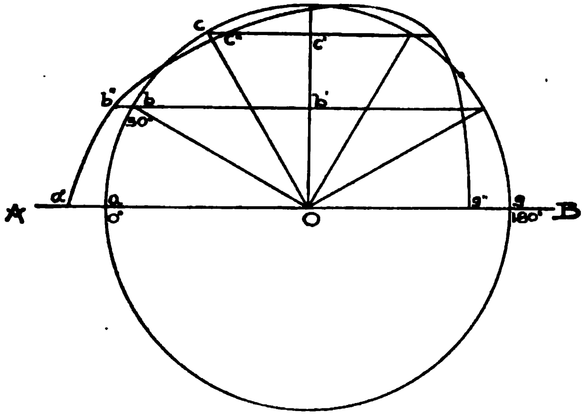


Fig. 102.

$0.00034WN^2r$, the rest of the curve being the cosine curve $a'b'c'$, etc., of this amplitude.

When

$$\begin{array}{ccccc} \theta = 0^\circ & \theta = 90^\circ & \theta = 180^\circ & \theta = 270^\circ & \theta = 360^\circ \\ 2\,\theta = 0^\circ & 2\,\theta = 180^\circ & 2\,\theta = 360^\circ & 2\,\theta = 540^\circ & 2\,\theta = 720^\circ \end{array}$$

$\cos 2\,\theta = \pm 1$; hence at d lay off dd'' downward and equal to

$$\frac{r}{l} \times 0.00034WN^2r.$$

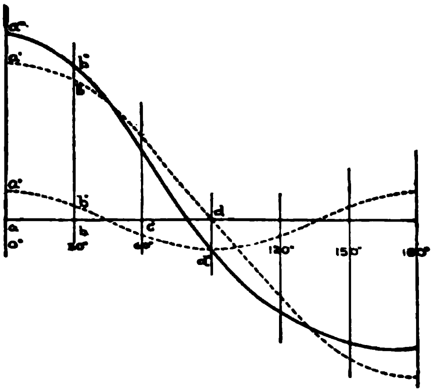


Fig. 103.

This gives the amplitude of the second-period cosine curve $a''b''c''$, etc. Combining these ordinates, there results the total curve *a'''b'''c'''d'''*, etc., giving the axial components of the inertia forces. The ordinates *aa'''*, *bb'''*, etc., of this curve are the same as those of the inertia diagram drawn to the piston-path base. Thus in Fig. 104, representing the usual inertia curve *a'g'*, is the piston path *a'b'*; *c'd'*, etc., are the piston position corresponding to the crank position of *abcd*, etc., of Fig. 103, while the ordinates *a*, *a'''* are precisely the same as in Figs. 103 and 102.

The effectiveness of the balance for the reciprocation inertia forces by the use of a revolving counterbalance is shown by combining the diagrams on an equal angle base for the two cases.

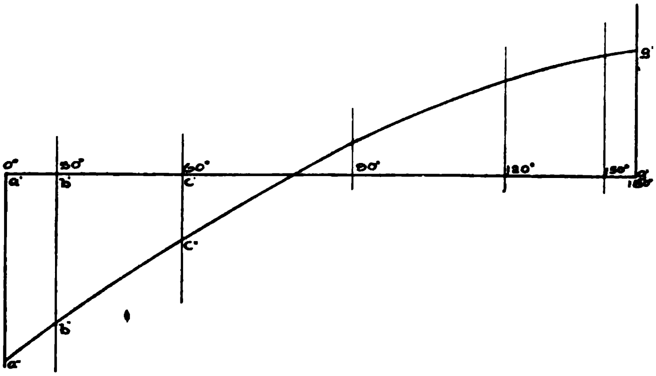


Fig. 104.

Thus for the axial components when there is no balance weight there will be two first-period cosine curves added to a second-period curve. One first-period cosine curve shows the axial components of the centrifugal force, while the other first-period and one second-period curve show the reciprocating effect.

The addition of a counterbalance weight introduces a first-period cosine curve of amplitude according to its mass and with a phase difference of 180°.

Should the amplitude of the cosine curve for the counterbalance be equal to the sum of the other two first-period curves, or, in short, should the mass of this counterbalance equal the mass of the reciprocating and revolving masses, then the unbalanced axial force is simply that of the second period.

This, however, greatly increases the transverse components, as has already been pointed out; but then the form is a sine curve.

Thus for masses selected as above the axial unbalanced force is a second-period cosine curve of amplitude $\frac{r}{l}\times$ that of the transverse unbalanced force of first-period and amplitude, $0.00034WN^2r$.

When several cylinders with various reciprocating and revolving parts are combined, then the unbalanced forces are given by the algebraic sum of the individual curve ordinates, but each must be first set out according to phase of cranks or counterbalances, as the case may be. By this means not only may the extent of the shaking forces be found, but the nature of their variation throughout one or several strokes and their periods.

The knowledge of the period of fluctuation is important in connection with possible synchronism with some of the framed supports, especially in boats, automobiles, and in buildings, particularly large, tall steel structures.

The tendency to rock or tilt depends, as has been noted, not on forces alone, but on the arm of the couples set up as well. These arms in an engine are all constant and it is only the forces which vary. The nature and extent of the variation can be found as shown. The couple curve might be drawn if it is desired; its form will be the same as that of the unbalanced force. Its amplitude for a system of numerous couples is to be found by the maximum closure of the couple polygon, all forces having arms with reference to an imaginary reference plane.

All the preceding work presupposes the connecting-rod to give effects the same as would result from two separate masses, one at the crank-pin revolving and the other reciprocating with the piston. A real rod is continuous, and its effects are different than those which result from the divided rod.

57. **Effect of Correctly Treating the Connecting-rod as a Continuous Mass Rather than a Divided Mass.**—Mr. John Macalpine has given a very clear statement of this in a paper in "Engineering," Oct. 22, 1897. He says:

"The true movement of the rod makes it necessary to investigate four problems.

"*A*. The inertia forces parallel to the axis of the cylinder.

"*B*. The inertia forces at right angles to the axis of the cylinder.

"*C*. The couple acting on the rod to produce its angular motion.

"*D.* The turning moment at the crank due to the inertia of the connecting-rod."

Investigating these effects the following conclusions are presented.

A. This force "*A*" can be expressed in a Fournier series of periods similar to that found for the reciprocating parts. For the first period the force *A* is not affected by the angularity of the rod, but there are terms of higher period which are not large or serious in general. The expression is

$$A=(0.00034WN^2r\left(\cos\ \theta+(1-k)\left\{\begin{array}{l}\cos 2\ \theta(a+\frac{1}{4}a^3+\frac{15}{128}a^5+\\-\cos 4\ \theta(\quad\frac{1}{4}a^3+\frac{3}{16}a^5+\\+\cos 6\ \theta(\qquad\quad+\frac{9}{128}a^5+\end{array}\right\}\right),$$

where $R=\frac{d}{l}=\frac{\text{distance from wrist-pin to rod center of gravity}}{\text{length of rod}}$.

B. This force at right angles to the cylinder is the same as if the mass of the rod were divided and a part concentrated at the crank-pin, inversely proportional to the distance of the center of gravity from the ends, and is given by

$$B=-(0.00034WN^2r)K\sin\theta.$$

C. The value of the couple is, if I = moment of inertia of rod about its center of gravity,

$$C=-0.00034N^2I\left(\begin{array}{l}\sin\ \theta(a+\frac{1}{8}a^3+\frac{3}{64}a^5+\text{etc.}\\+\sin 3\ \theta(+\frac{3}{8}a^3+\frac{27}{128}a^5+\text{etc.}\\-\sin 5\ \theta(\qquad\quad+\frac{15}{128}a^5+\text{etc.}\end{array}\right).$$

D. The turning moment due to the rod of weight *W* is

$$D=0.00034WN^2r^2(1-k)\left\{\begin{array}{l}\sin\theta(\frac{1}{4}a+\frac{1}{16}a^3+\text{etc.}\\-\sin 2\ \theta\times\frac{1}{2}(1+K)\\-\sin 3\ \theta(\frac{3}{4}a+\frac{9}{32}a^3+\text{etc.}\\-\sin 4\ \theta(\times\frac{1}{4}a^2(1=K)\\+\sin 5\ \theta(\frac{5}{32}a^3+\text{etc.}\end{array}\right\}\begin{array}{l}+\frac{a^2Iw^2}{2g}\\ \sin 2\ \theta\\+\text{etc.},\end{array}$$

which, when $K=0$ and $I=0$, reduces to the expression for mass at crosshead. The magnitude of these effects—that is, the magnitude of the variation introduced by the correct effects compared with approximations resulting from dividing the rod—are of no consequence so far as turning effort is concerned, but may be in special cases where fine balancing is required. No accurate calculation of free forces tending to shake rock or vibrate the bed can be made without considering these exact conditions, but in most gas-engines up to the present time, balance, where secured, has been attained by arrangement of counter parts as explained, without knowing just what was the rod effect. Should future work demand such analysis, designers may refer to the work on the subject by Professors Klein, Dunkerly, and Dalby, Messrs. Macalpine, Macfarlane, Grey, Admiral Melville, and others, the best of which have appeared in the proceedings of the societies of naval engineers.

58. Turning Effort Corrected for Inertia (Approximately) and Divided Rod.—The force acting on the wrist-pin is transmitted to the crank through the oblique connecting-rod. The tangential component of this transmitted force, or so much as is transmitted, is the turning force and the diagram of turning forces showing the changes in the tangential forces is the crank-effort diagram. When multiplied by the crank radius it gives the turning moment. The area of the crank-effort diagram on the developed crank *circle* as a base gives the work *done per cycle* and should equal in area the indicator card. Its mean ordinate is the mean turning effort or, as more generally called, the mean resistance. The tangential turning force is found either algebraically or graphically. Graphically the effective pressure P, Fig. 105, is laid off on AB and resolved by parallelogram into components BD against the slide and DA along the rod. $D'A' = DA$ at crank is similarly resolved into normal component $D'F$, exciting a force through the crank on main bearing

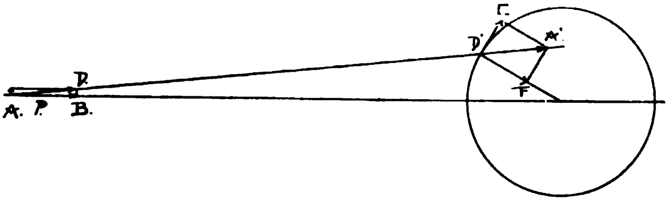


Fig. 105.

and *D'E*, the tangential force. The turning effort, or tangential force, can be found in terms of the effective pressures for all crank-angles and can then be most easily plotted directly in connection with the tables given for inertia and piston positions. Such a table prepared by Klein is here given.

59. Rotative Effect at Crank of Unit Horizontal Pressure at Wrist-pin.—To find tangential force multiply tabular quantity by effective pressure at wrist-pin.

Crank Angles.		Rod + crank $= \frac{1}{a_1}$.				
Forward.	Return.	4.0	4.5	5.0	5.5	6.0
0	80	0	0	0	0	0
30	150	.6091	.5968	.5870	.5791	.5724
45	135	.8341	.8195	.8081	.7988	.7910
60	120	.9769	.9641	.9540	.9458	.9390
90	90	1.0000	1.0000	1.0000	1.0000	1.0000
120	60	.7552	.7680	.7781	.7863	.7931
135	45	.5801	.5946	.6061	.6155	.6232
150	30	.3909	.4032	.4130	.4209	.4276
180	0	0	0	0	0	0

These are obtained from the relations

$$t = \text{tangential force at crank-pin},$$

$$t = P_e \frac{\sin(\theta+\phi)}{\cos\phi};$$

in which

$$\sin\phi = a_1 \sin\theta,$$

$$\cos\phi = \sqrt{1 - a_1^2 \sin^2\theta}.$$

The graphic method is not so often used as the method involving the use of tables because the latter is quicker and more accurate. The method by tables may be followed in two ways, the first involving a determination of all pressures for equal crank-angles and the second the same for equal divisions of piston path.

60. First Method of Equal Crank-angles.—Prepare a table with the following data, starting with crank-angles.

1°. Find corresponding piston position for column 2 by the table.

2°. Find inertia factor which is the value of $(\cos\theta + a_1 \cos 2\theta)$ from the table and insert for column 3.

3°. Multiply the inertia factor by the engine constant, $0.00034\, W/a N^2 r$ and place the products in column 4.

4°. Draw on the indicator card vertical lines corresponding to the crank-angles as given for piston position in column 2 and measure the gas pressure by a scale for every piston position during four strokes. This will give columns 5, 6, 7, and 8.

5°. Compare inertia forces and gas forces and observe whether driving or resisting, this will give net forces at the piston for columns 9, 10, 11, and 12.

6°. Find the tangential force in terms of piston force for each crank-angle for column 13, the tangential factor.

7°. Multiply the piston force, net, by the tangential factor and insert in columns 14, 15, 16, and 17.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
				Gas Pressures.				Net Piston Pressures.					Tangential Pressures.			
Angle.	Piston Position.	Inertia Factor.	Inertia Force.	Expansion.	Exhaust.	Suction.	Compression.	Expansion.	Exhaust.	Suction.	Compression.	Tangential Factor.	Expansion.	Exhaust.	Suction.	Compression.
0																
30°																
45°																
60°																
90°																
120°																
135°																
150°																
180°																

The results in the last four columns should be plotted as ordinates to the first column, as abscissæ repeated four times, equal spaces indicating equal angles and the whole base line being twice the circumference of a circle whose diameter is the stroke.

61. Second Method of Equally Divided Piston Travel.—Having drawn the inertia diagram and indicator card to the same scale, Fig. 106, and on cross-section paper, divided into tenths so that fractions of the stroke as low as $0.02r$ can be read off, measure ordinates for each piston position and by scale or graphically combine them. For a four-cycle engine it is helpful for clearness to lay down on a four-stroke base the gas-pressure diagram of Fig. 107,

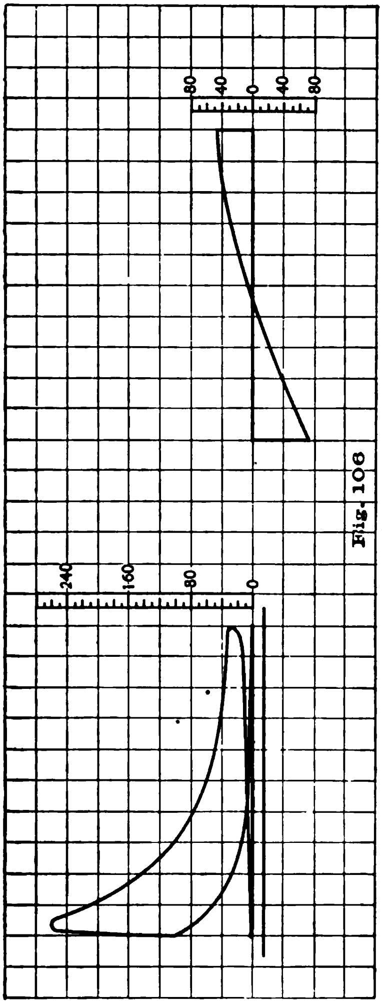


Fig. 106

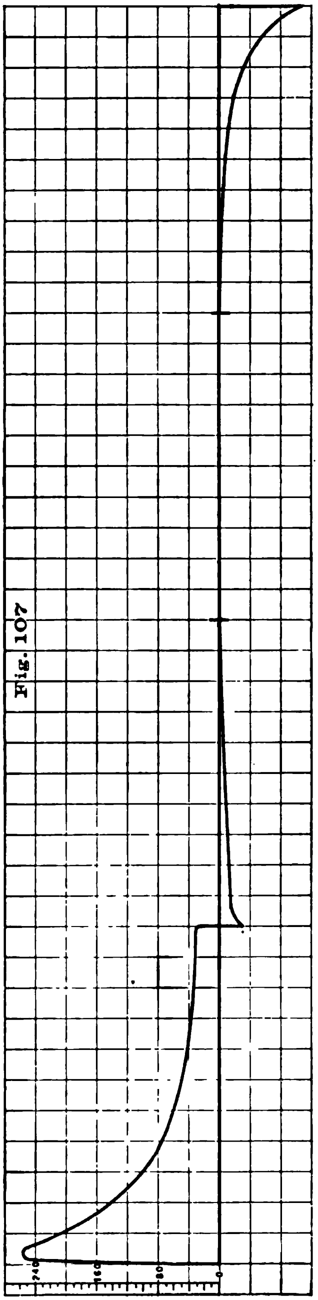


Fig. 107

beginning with expansion and taking the phases in the following order—exhaust, suction, compression. Combining with the inertia diagram gives the effective or net pressure curve at the piston (Fig. 108). Placing these results in a table and using the tangential factor will give the tangential force for each piston position. This latter, by the table, can be transformed to the tangential force for proper angles corresponding to piston position. For this case with a card on natural gas and maximum initial inertia of 71 lbs. per square inch the table found follows.

EXPANSION STROKE.

Fraction of Piston Travel.	Gas Pressure.	Inertia Pressure.	Net Pressure.	Tangential Factor.	Tangential Force.	Crank Angle.
.00	99	−71	28	0	....	0
.02	256	−68	188	.2220	....	14.70
.04	259	−64	195	.4277	....	21.14
.06	251	−61	190	.5170	....	26.00
.10	207	−55	152	.6185	....	33.80
.20	141	−38	103	.8544	....	49.98
.30	101	−23	78	.9654	....	61.89
.40	75	− 8	67	1.0075	....	73.00
.50	59	6	65	1.0146	....	84.40
.60	48	18	66	.9628	....	97.75
.70	40	29	69	.8900	....	108.00
.80	36	38	74	.7666	....	121.82
.90	33	44	77	.5533	....	139.22
.98	32	46	78	.2501	....	162.00
1.00	25	47	72	0	....	180.00

These turning forces or tangential pressures may be plotted above a base line equal in length to that used for the piston-pressure diagrams, in which case the scale is $\frac{2}{\pi}$ (horizontally) times the scale of the former, since the length, to have the scales equal, should be $\frac{\pi}{2}$ times length of piston-pressure diagram. This gives for this case Fig. 109. As the mean effective pressure for the indicator card is for one stroke, the mean driving force for four strokes, or two revolutions, is $\frac{P}{4}$. Calling mean turning force F since the same work must be done at crank-pin as at piston,

$$\frac{P}{4} \times 4L = F' \times 2\pi L,$$

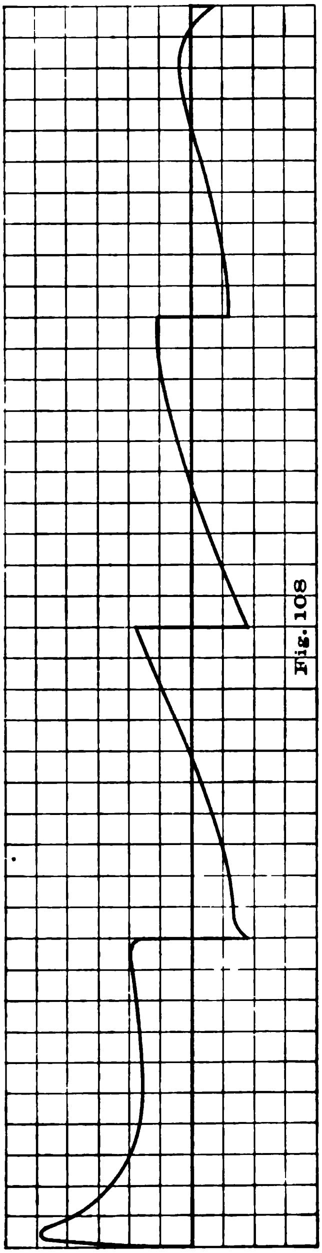


Fig. 108

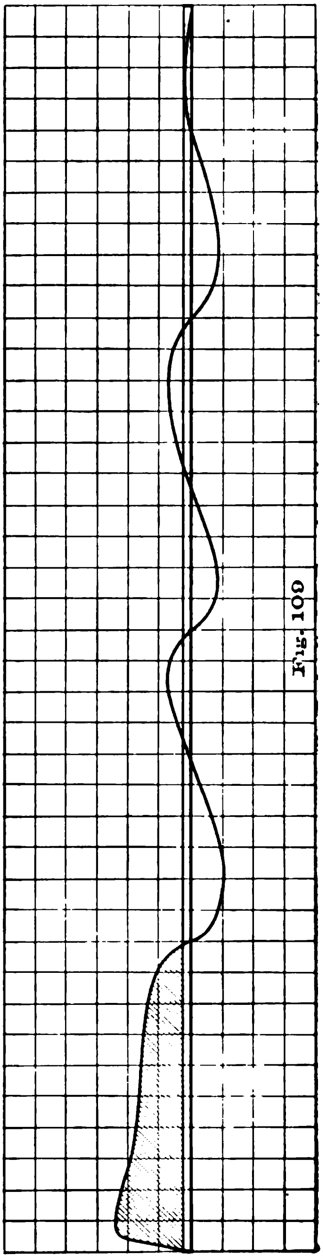


Fig. 109

or $$F_t = \frac{P}{2\pi}. \qquad (74)$$

This mean turning force is indicated on the turning-effort diagram of Fig. 109.

62. **Data from Turning-effort Diagram.** — Having found the turning-effort diagram, data that are exceedingly useful to the designer can be read off by inspection. The most important facts to be thus learned are:

1°. Relation between maximum turning force and mean turning force to assist in finding maximum turning force when for a new engine only the mean is given. This is available at once in crank-shaft design.

2°. The maximum energy ΔE delivered by the engine over the mean E of the cycle for any given case. In Fig. 109 this is the ratio of shaded area to the long narrow mean area. This is a measure of the fly-wheel weight necessary to limit angular velocity variations during the cycle.

3°. The number of times the turning force changes direction; that is, the number of times the real curve crosses the mean. This is a measure of the period of the rocking force of the frame as a whole about the crank-shaft. It is also the number of knocks that would be heard on a loose pin.

It is necessary to know how these three results and other related data will vary with conditions, and for this purpose a great number of turning-effort diagrams have been drawn, some of which will be presented. The conditions which might affect turning effort are:

1°. Kind of gas.
2°. Method of working (two-cycle or four).
3°. Combination of cylinders, phases, and crank-angles.
4°. Inertia very low or very high.
5°. Speed.
6°. Load.
7°. Mixture.
8°. Ignition and throttle.
9°. Size of engine.

63. **Change of Turning Effort with Conditions.**—The case given represents a medium power engine on natural gas and with average inertia giving for a single-cylinder four-cycle engine:

$P = 64.48$ lbs. per sq. in.,
$F_t = 10.26$ " " " "

$$\frac{\text{Maximum turning force}}{\text{Mean turning force}} = 9.45,$$

$$\frac{\Delta E}{E} = 1.31.$$

The line crosses the mean approximately at every dead-center, and for the idle strokes also about mid-stroke. A little greater inertia would also have caused a reversal toward the end of the compression stroke. A change of load would have done the same if compression were lowered thereby. A smaller engine on carbureted water-gas having a lower inertia, 22 lbs. initial, gave the results of Fig. 110, in which (*a*) is the indicator card, (*b*) the inertia diagram, (*c*) the net driving-pressure curve at the piston. The turning-effort curve, Fig. 111, shows, because of the lower inertia, fewer reversals of effort, which is positive approximately for one stroke and negative for three.

For this case

$P = 82.5$ lbs. per sq. in. of piston,
$F_t = 13.12$ " " " " " "

$$\frac{\text{Maximum turning force}}{\text{Mean turning force}} = 7.77,$$

$$\frac{\Delta E}{E} = 1.17.$$

The last case was for a full-load card on best adjustment. When the card becomes flat through bad mixture or ignition, evidently the forces vary. A poor-mixture card is shown in Fig. 112, (*a*), (*b*), (*c*), with inertia as before, and the turning effort is given in Fig. 113.

For this case

$$P = 43.8,$$

$$F_t = 6.97,$$

$$\frac{\text{Maximum turning force}}{\text{Mean turning force}} = 12.12,$$

$$\frac{\Delta E}{E} = 1.53.$$

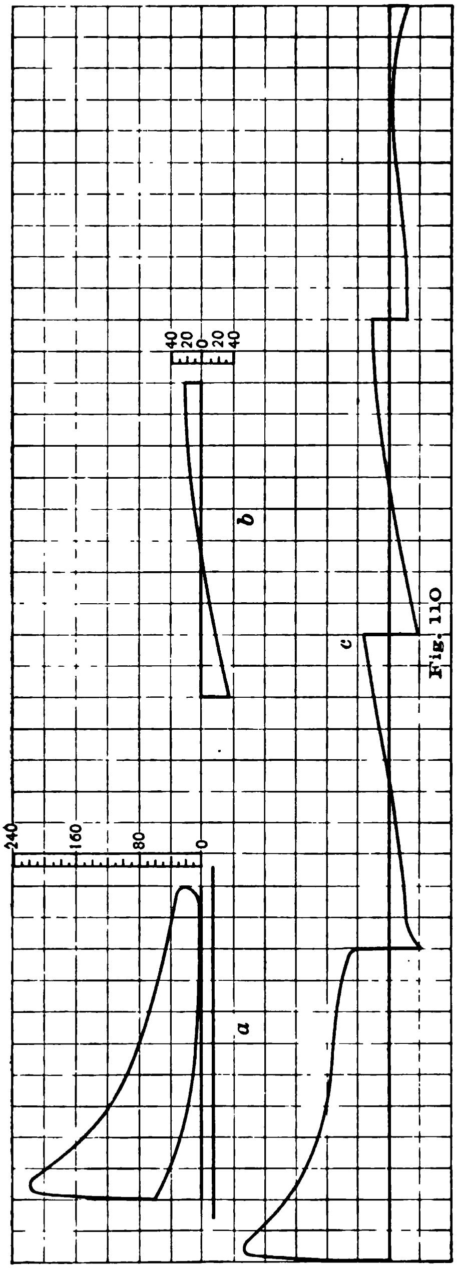


Fig. 110

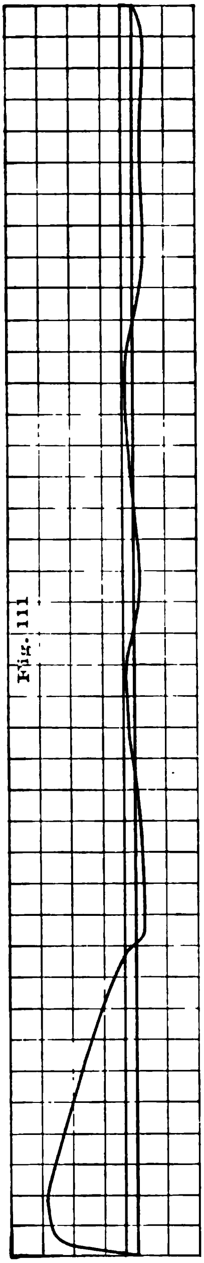


Fig. 111

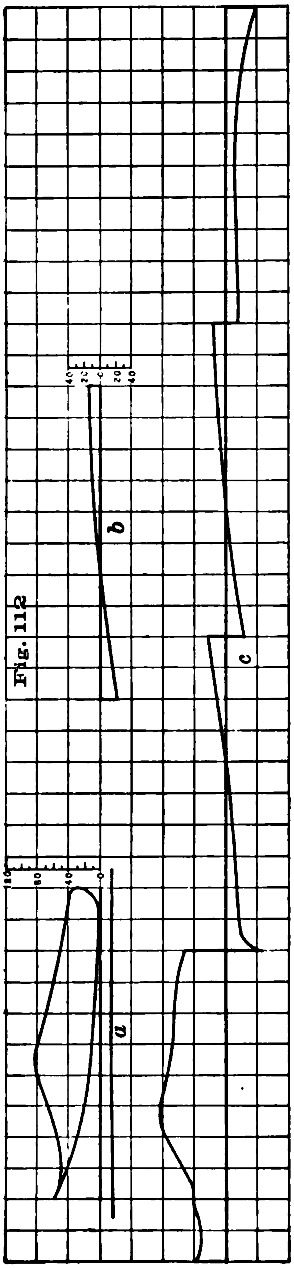


Fig. 112

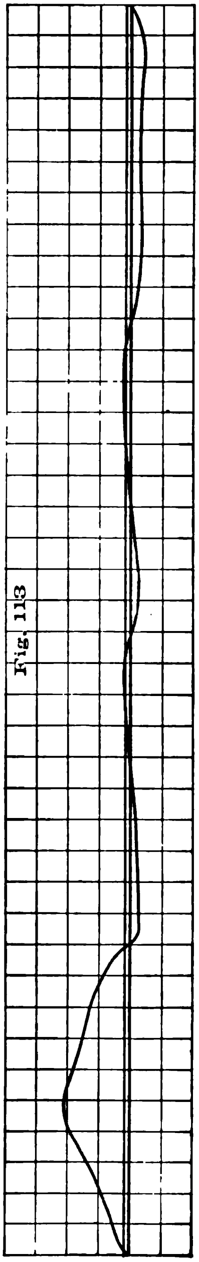


Fig. 113

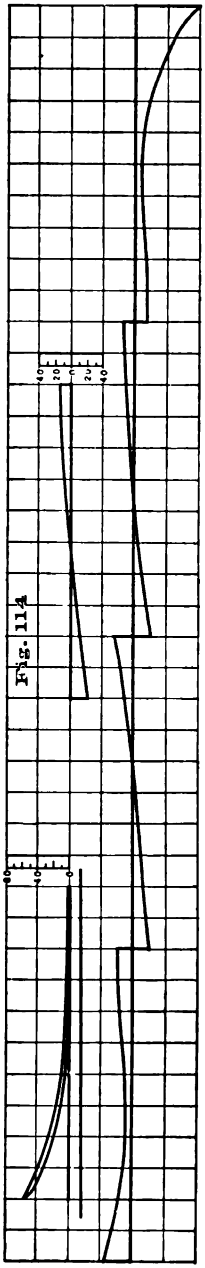


Fig. 114

It is hardly fair and not instructive to compare maximum turning force to the new light-load mean, as is here done. It is much better to compare with the full-load mean. This gives

$$\frac{\text{Maximum turning force (bad mixture)}}{\text{Mean turning (force full load)}} = 6.43.$$

Similarly for the ΔE the comparison when made with the full-load mean E gives

$$\frac{\text{Poor mixture } \Delta E}{\text{Full load } E} = 0.81.$$

The force of this appears when fly-wheels are to be designed. It shows that for this poor-mixture card there will be less angular velocity change than for the full-load card.

Governing by the hit-or-miss method introduces idle strokes in which the compression curve is approximately repeated on expansion, as in Fig. 114, Fig. 115 giving the corresponding turning effort. This is practically the no-load case for the engine giving the cards Fig. 110 and Fig. 112. The mean effort is practically zero, and therefore when a full-load cycle is followed by one cycle like this, the ratios given above are to be doubled.

A moderately small engine on kerosene with a maximum initial inertia of 18½ lbs. gave the results shown in Fig. 116, where (*a*) is the indicator card, (*b*) the inertia, and (*c*) the curve of net piston pressures. Here the inertia forces are insignificant compared with the gas-pressure forces. The turning-effort curve, Fig. 117, as a consequence has but one pair of reversals, the engine being accelerated one stroke, nearly, and meeting continuous resistance for three.

$$P = 92.8,$$

$$F_t = 14.78,$$

$$\frac{\text{Maximum turning force}}{\text{Mean turning force}} = 8.28,$$

$$\frac{\Delta E}{E} = 1.07.$$

A producer-gas card and turning-effort curve with a little larger initial inertia, 49 lbs., is shown in Fig. 118. The curve has the

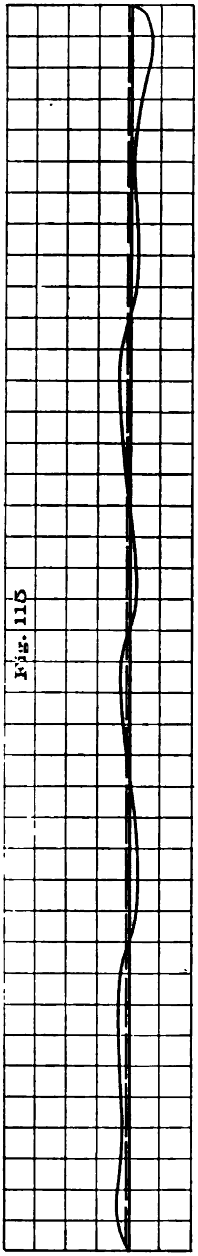


Fig. 115

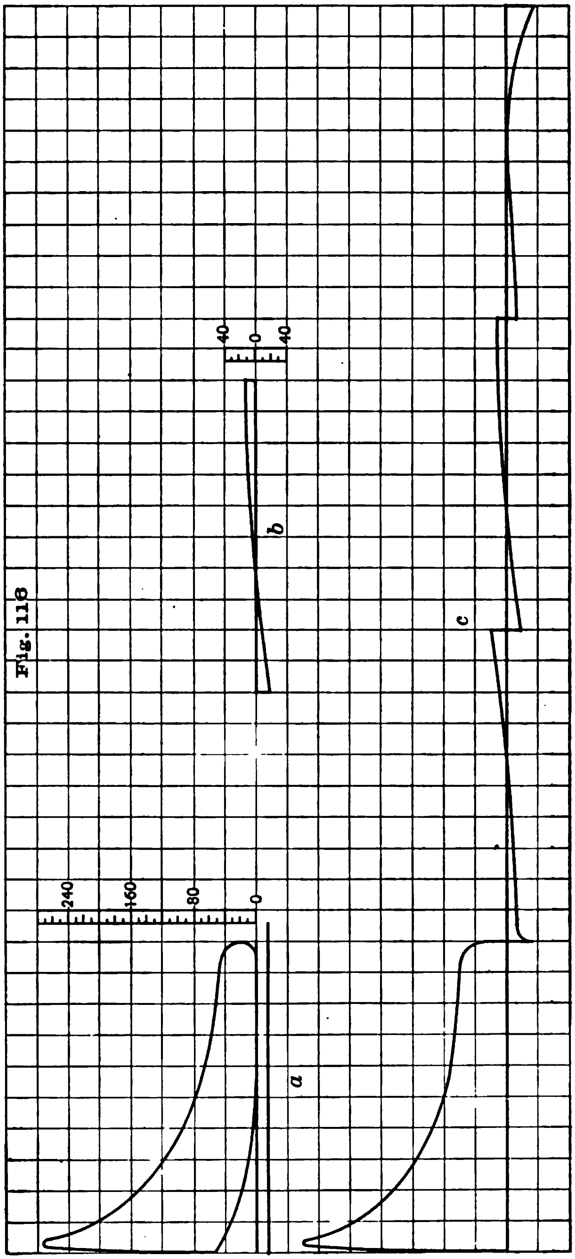


Fig. 116

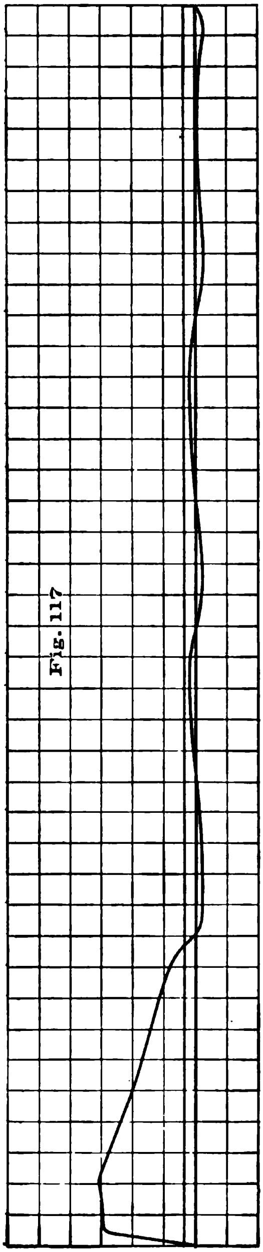


Fig. 117

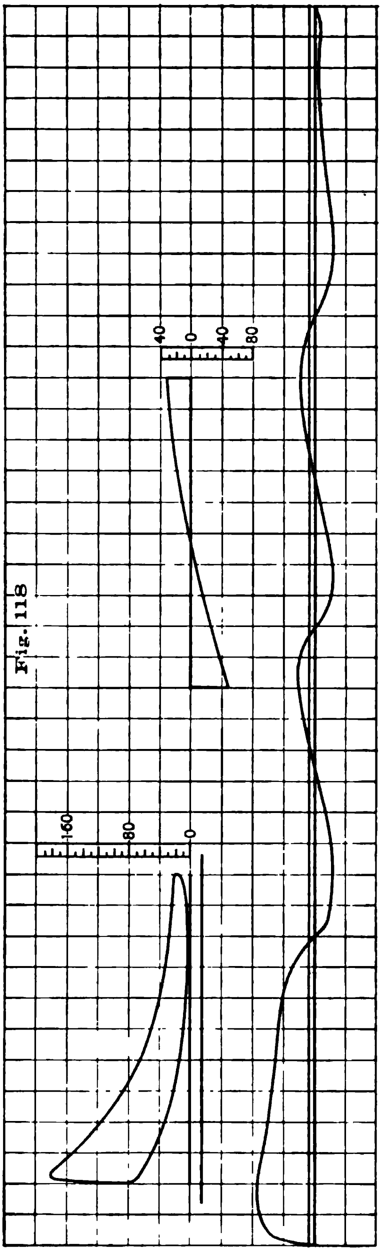


Fig. 118

characteristic wave for high inertias, though this value is not high except when compared with the mean turning force. For this case

$$P = 48.64,$$

$$F_t = 7.68,$$

$$\frac{\text{Maximum turning force}}{\text{Mean turning force}} = 9.50,$$

$$\frac{\Delta E}{E} = 1.41.$$

64. Speed Effect on Turning Effort.—Speed increase may affect a card by flattening when the rate of propagation in the mixture is constant, at the same time inertia forces increase. Therefore the effect on the turning-effort diagram will be double as both curves change. This is illustrated in the following two cases. A card from a small gasoline-engine, Fig. 119, at slow speed with only one pound initial inertia, as the engine was very small, gave the turning-effort curve shown. This curve gives the data following:

$$P = 48.8,$$

$$F_t = 7.77,$$

$$\frac{\text{Maximum turning force}}{\text{Mean turning force}} = 9.14,$$

$$\frac{\Delta E}{E} = 1.14.$$

On increase of speed, so as to increase inertia to an initial value of 35 lbs. per square inch by speed increase alone, the turning effort of Fig. 120 results. Here the nature of the curve has become quite different not only during the working stroke but also during the rest of the cycle:

$$P = 28.96,$$

$$F_t = 4.61,$$

$$\frac{\text{Maximum turning force}}{\text{Mean turning force}} = 11.91,$$

$$\frac{\text{Maximum turning force (reduced card)}}{\text{Mean turning force (original card)}} = 7.2,$$

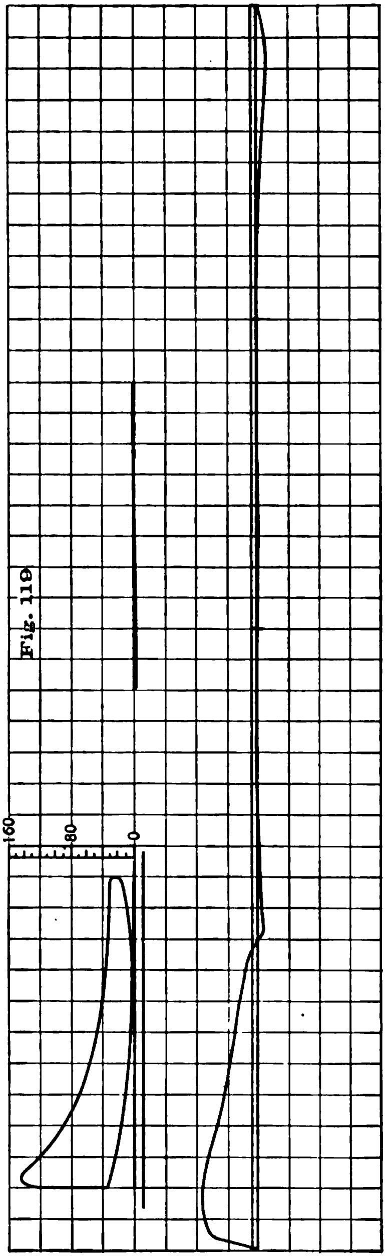


Fig. 119

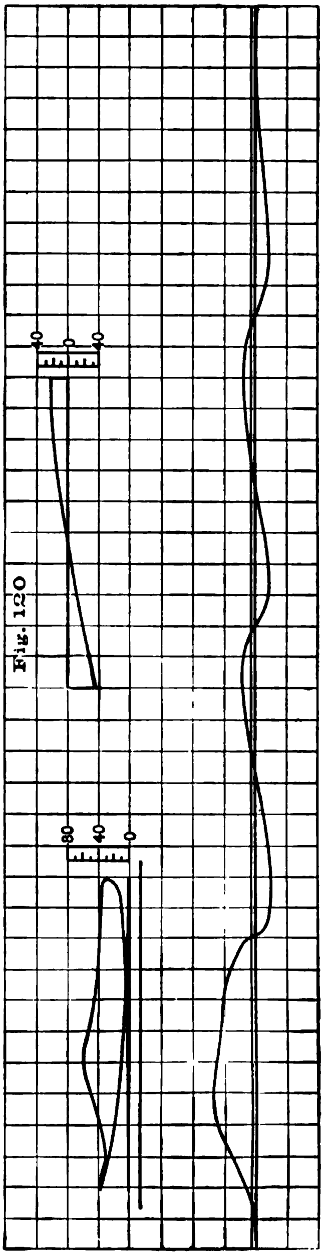


Fig. 120

$$\frac{\Delta E}{E}=1.43,$$

$$\frac{\Delta E}{\text{Original } E}=0.85.$$

The last case of speed change involved a combination of increasing inertia forces and decreasing gas pressures. When, however, an automobile engine is carrying a fairly steady load resistance, speed may be varied by throttle and ignition changes, and these effects are shown in the succeeding series of curves, in which the speed has been taken proportional to the mean pressure P and the highest value is $N=2052$. This case is shown in Fig. 121, for which

$$P=75.52,$$

$$F_t=12.02,$$

$$\frac{\text{Maximum turning force}}{\text{Mean turning force}}=11.27.$$

$$\frac{\Delta E}{E}=1.23.$$

By decreasing speed on this full-load card to 565 there results Fig. 122, for which P and F_t are the same, but the ratio of maximum to mean turning force has fallen to 6.74, proving a most important fact, that this ratio is dependent on inertia. Also the ratio $\frac{\Delta E}{E}$ has fallen by decrease of inertia only to 1.15 from 1.23 because of the redistribution of effort. Decreasing the speed by the throttle to the effects shown by Fig. 123, and still further by late ignition to Fig. 124, and finally by both actions simultaneously to Fig. 125, gives the resultant effects tabulated.

	Throttle Closure, Fig. 123.	Late Ignition, Fig. 124.	Close Throttle and Late Ignition, Fig. 125.
P	41.60	40.00	11.04
F_t	6.62	6.37	1.76
$\frac{\text{Maximum turning force}}{\text{Mean turning force}}$	7.17	13.50	14.77
$\frac{\Delta E}{E}$	1.10	1.68	2.25
Initial inertia	74.5	69.0	5.0

The conclusions are evident and in line with previous comments.

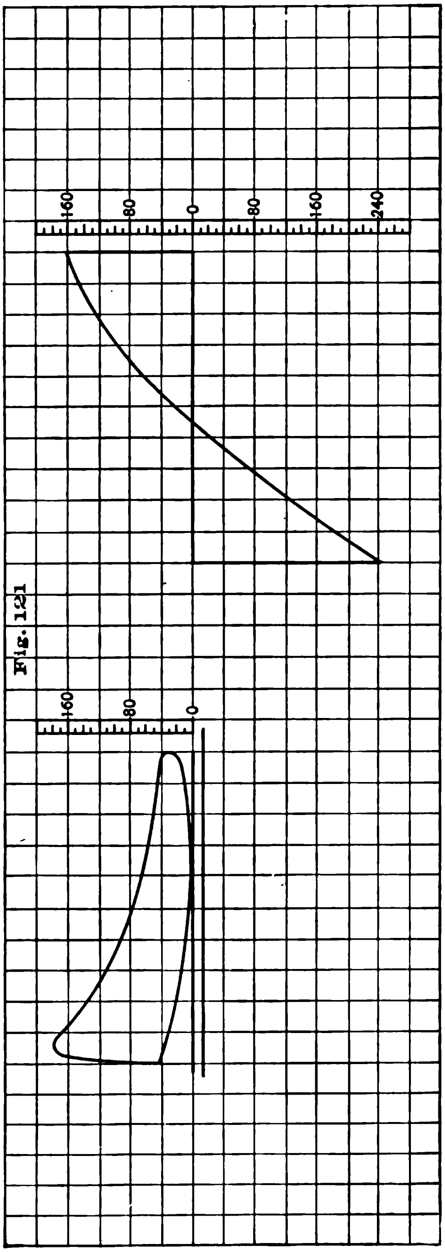


Fig. 121

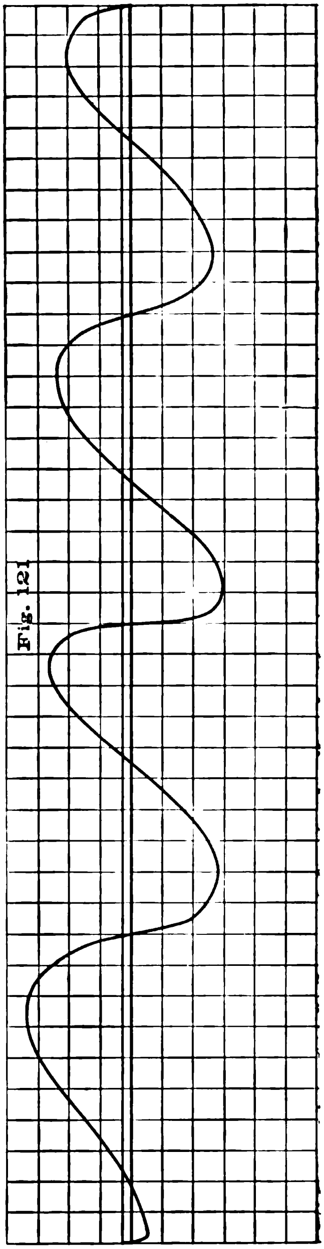


Fig. 121

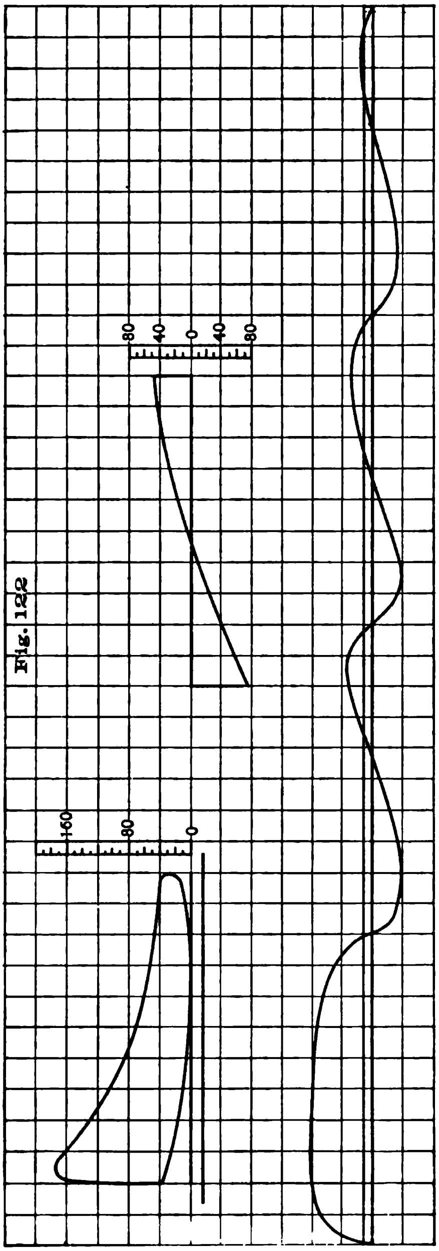


Fig. 122

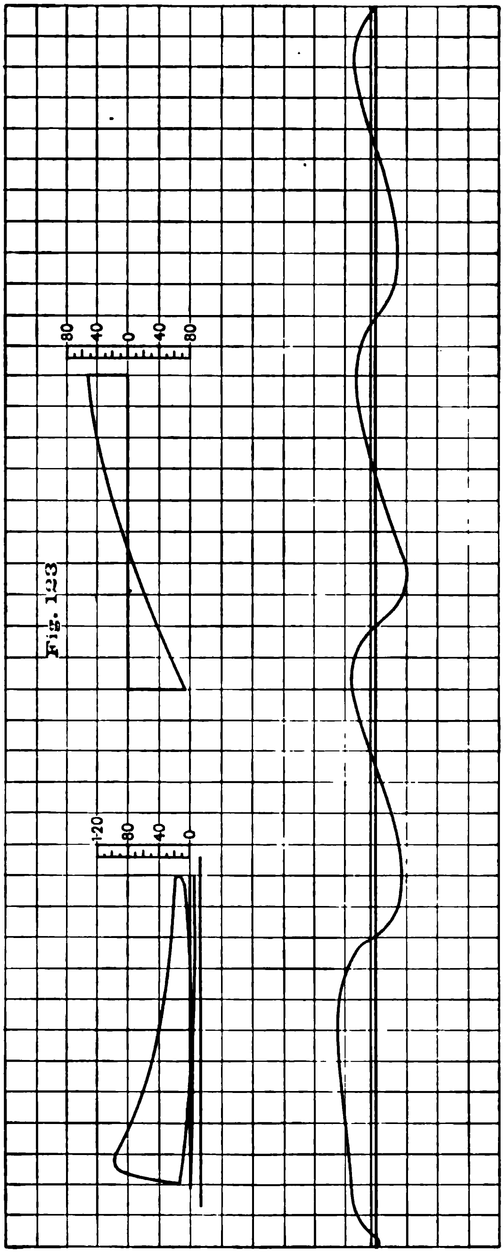


Fig. 123

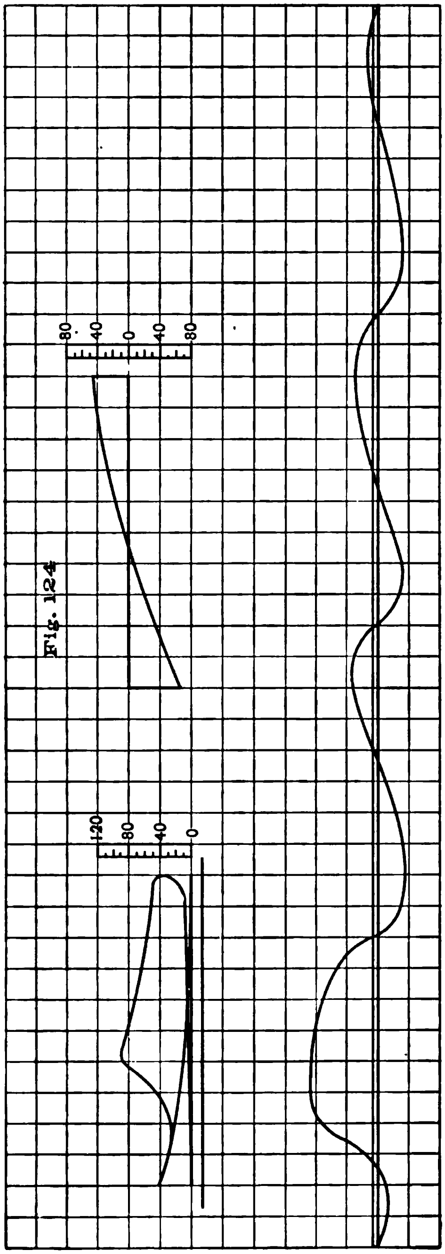


Fig. 124

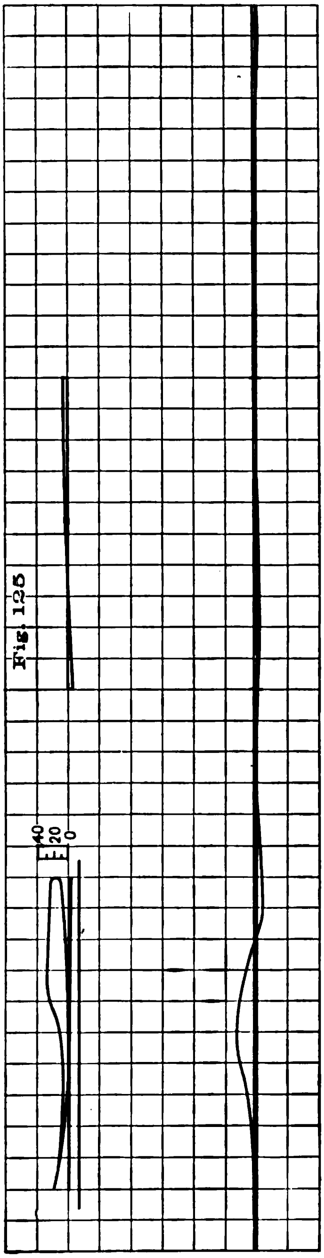


Fig. 125

65. Two-cycle Turning Effort.—Two-cycle engines pass through their complete series of pressure changes in two strokes, or one revolution, therefore their effective or net pressure curves and turning-effort diagrams are laid off on base lines only half as long as for four-cycle engines. A two-cycle card and its corresponding inertia diagram are shown in Fig. 126, (*a*) and (*b*); (*c*) represents the piston net pressures and (*d*) the turning-effort curve. In general, the mean turning effort is twice that for a four-cycle card and given by $F_t = \frac{P}{\pi}$.

For this curve the characteristics other than the simple single reversal of effort on each dead-center are:

$$P = 63.52,$$

$$F_t = 20.22,$$

$$\frac{\text{Maximum turning force}}{\text{Mean turning force}} = 4.20,$$

$$\frac{\Delta E}{E} = 1.26.$$

The fact that the mean tangential force is higher for a two-cycle and twice its amount for a four-cycle card has also the effect of halving the ratio of maximum to mean.

When cylinders are double-acting, the turning effort for one side must be combined with that for the other side. In making this combination there are four ways in which the four phases in two sets can be combined as follows, if four-cycle:

Head end	Compression	Expansion	Exhaust	Suction	(1)
Crank end	Suction	Compression	Expansion	Exhaust	
Head end	Expansion	Exhaust	Suction	Compression	(2)
Crank end	Suction	Compression	Expansion	Exhaust	
Head end	Exhaust	Suction	Compression	Expansion	(3)
Crank end	Suction	Compression	Expansion	Exhaust	
Head end	Suction	Compression	Expansion	Exhaust	(4)
Crank end	Suction	Compression	Expansion	Exhaust	

When, however, the cylinders are two-cycle the combination of phases is:

Head end	Compression	Expansion	(1)
Crank end	Compression	Expansion	
Head end	Compression	Expansion	(2)
Crank end	Expansion	Compression	

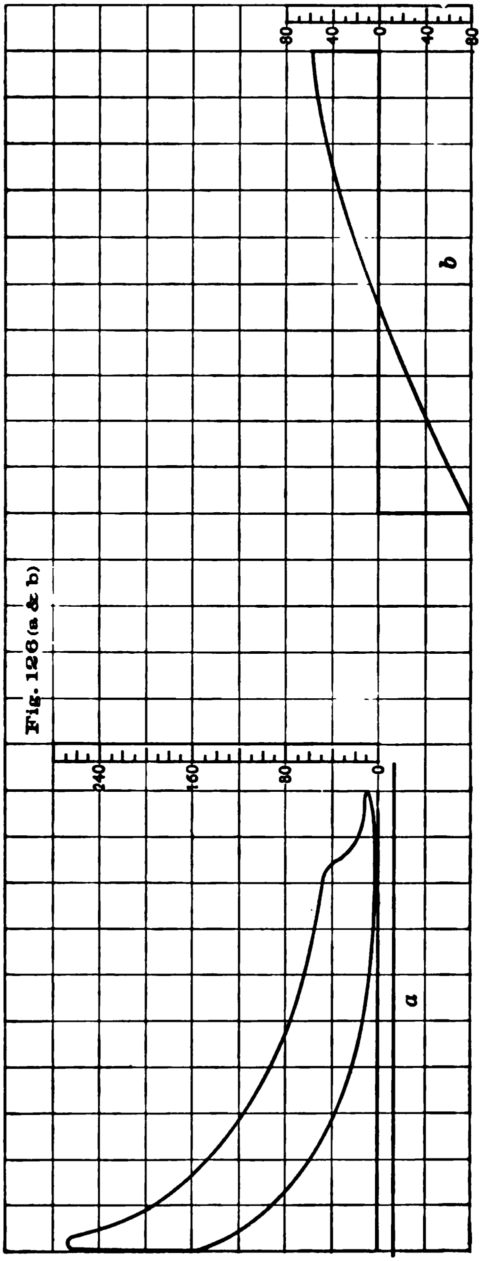


Fig. 126 (a & b)

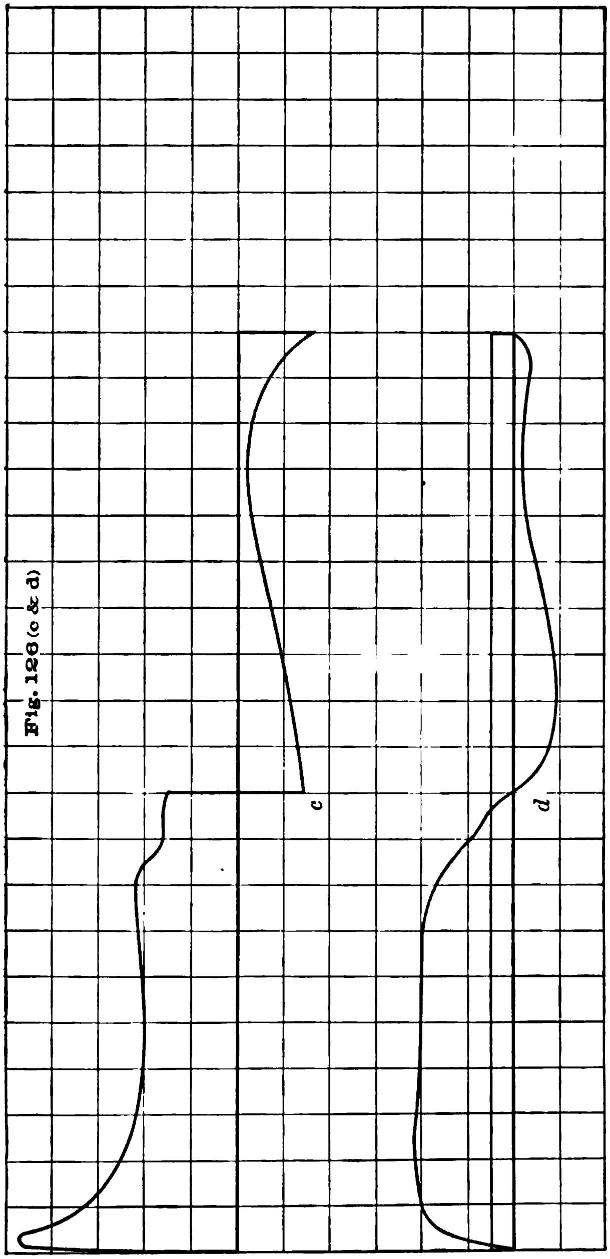


Fig. 126 (c & d)

As the result of combining by method (1) is simply to double ordinates, and not to nullify positive by negative force, the latter method (2) is preferable.

The two-cycle engines are often, in fact generally, fed from separate compressors which may have separate cranks and reciprocating parts. In cases like this the resisting-effort curve for the air-pumps must be combined, observing the appropriate angular lead, with the turning-effort diagram of the power cylinders. In this way Fig. 127 was drawn for a medium-powered, double-acting,

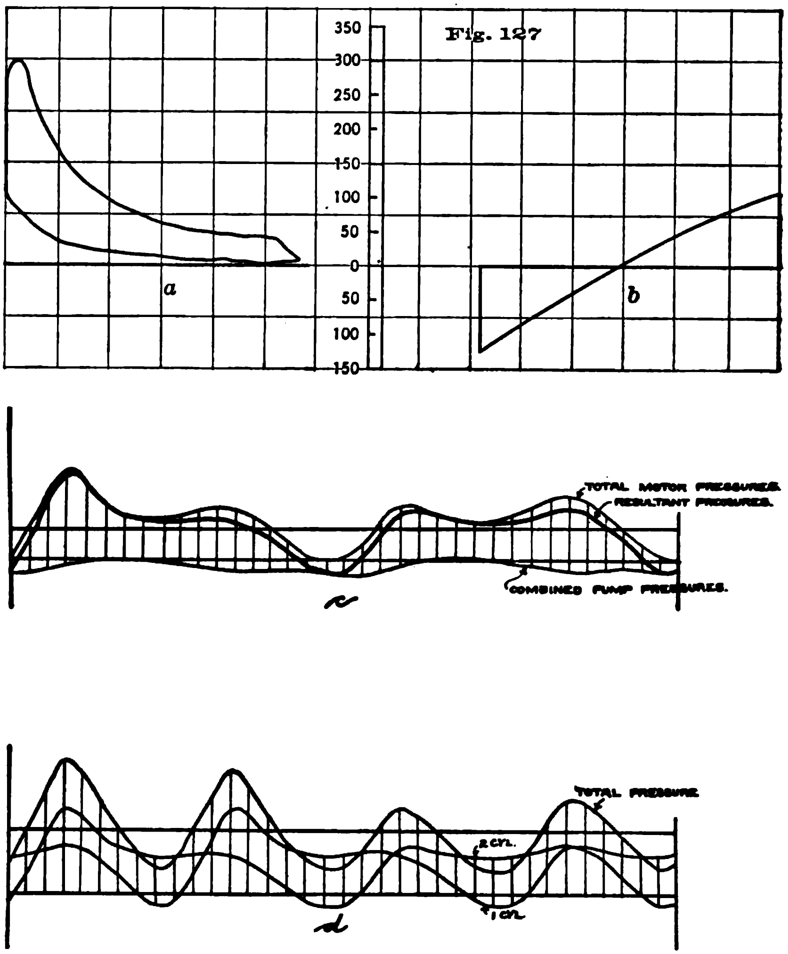


Fig. 127

single-cylinder, two-cycle engine. Fig. 127 (*a*) is the power indicator card, (*b*) the inertia diagram for the weights and speed, (*c*) the turning effort for motor cylinder, pump cylinder, and net turning effort for one cylinder, (*d*) the turning effort for two cylinders with proper phase difference of 90°, the crank-angle between these two double-acting cylinders. These two diagrams give the following data:

	Two-cycle Double-acting.	
	One Cylinder.	Two Cylinders.
P	78.5	78.5
F_t	25.0	50.0
$\frac{\text{Maximum turning force}}{\text{Mean turning force}}$	2.84	2.13
$\frac{\Delta E}{E}$	.38	.17

66. Phase Relations and Inertia on Turning Effort.—The two most important influences to be studied are the combination of phases and their effect on the above and the influence of inertia on the same result. For this purpose Fig. 128 (*a*), (*b*), (*c*), (*d*), etc., is presented in connection with Fig. 129 (*a*), (*b*), (*c*), (*d*), etc. Both sets are based on the same indicator card, (*a*) from an engine using producer-gas, but in each case the inertia (*b*) is different; in Fig. 128 it is unusually low, 26 lbs. initial, and in Fig. 129 unusally high, 208 lbs. initial. In both figures the letters have the following significance:

a = indicator card;

b = inertia diagram;

c = turning effort, one-cylinder;

d = " " two- " phases $\frac{720^\circ}{2} = 360^\circ$;

e = " " three- " " $\frac{720^\circ}{3} = 240^\circ$;

f = " " four- " " $\frac{720^\circ}{4} = 180^\circ$;

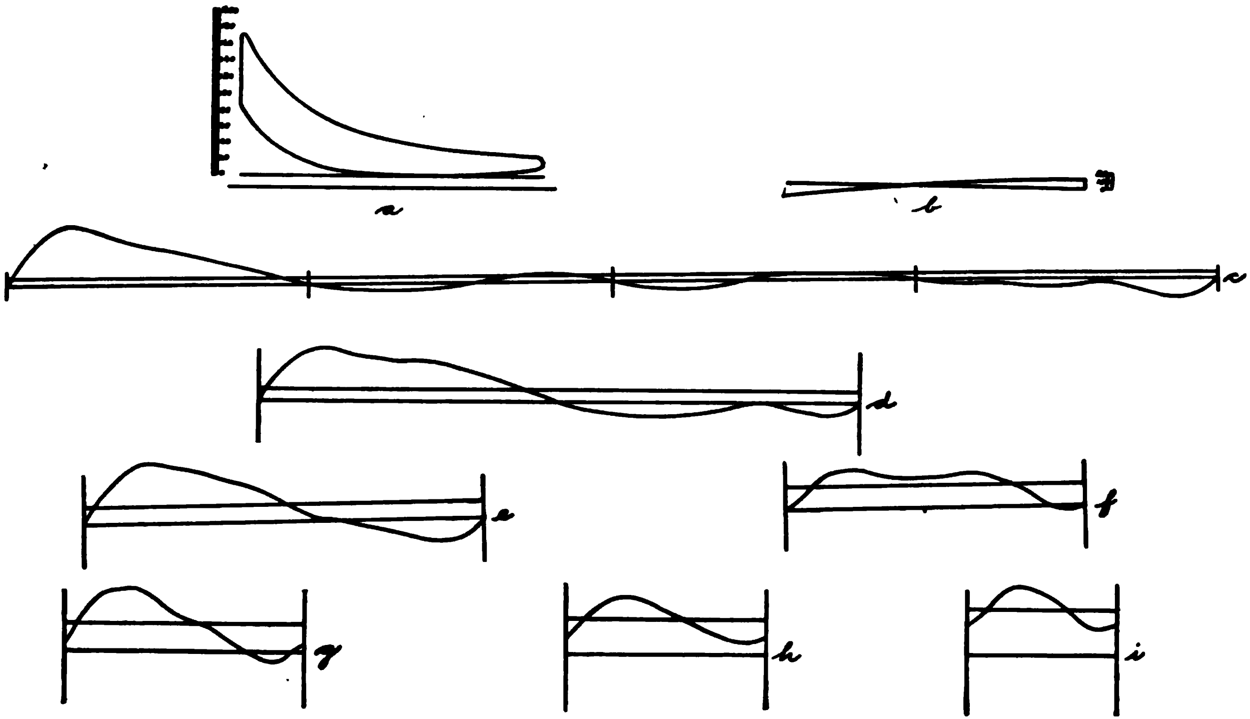


Fig. 128.

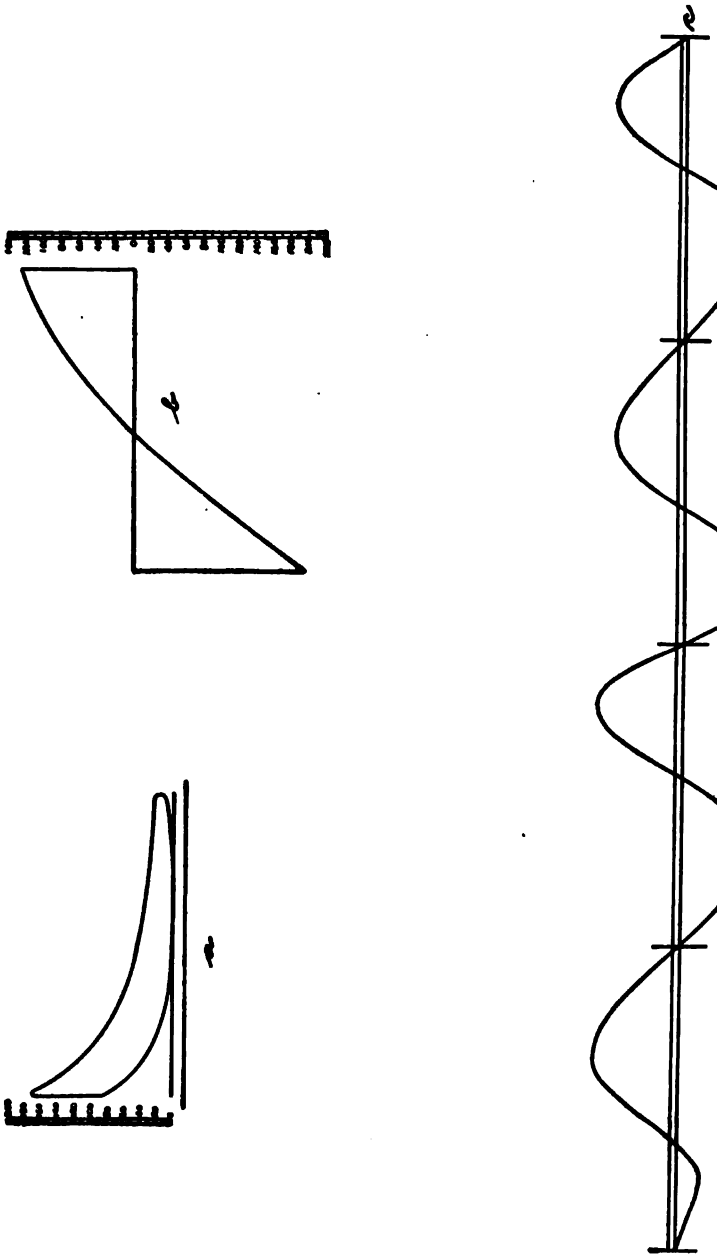


Fig. 129 (*a* to *c*).

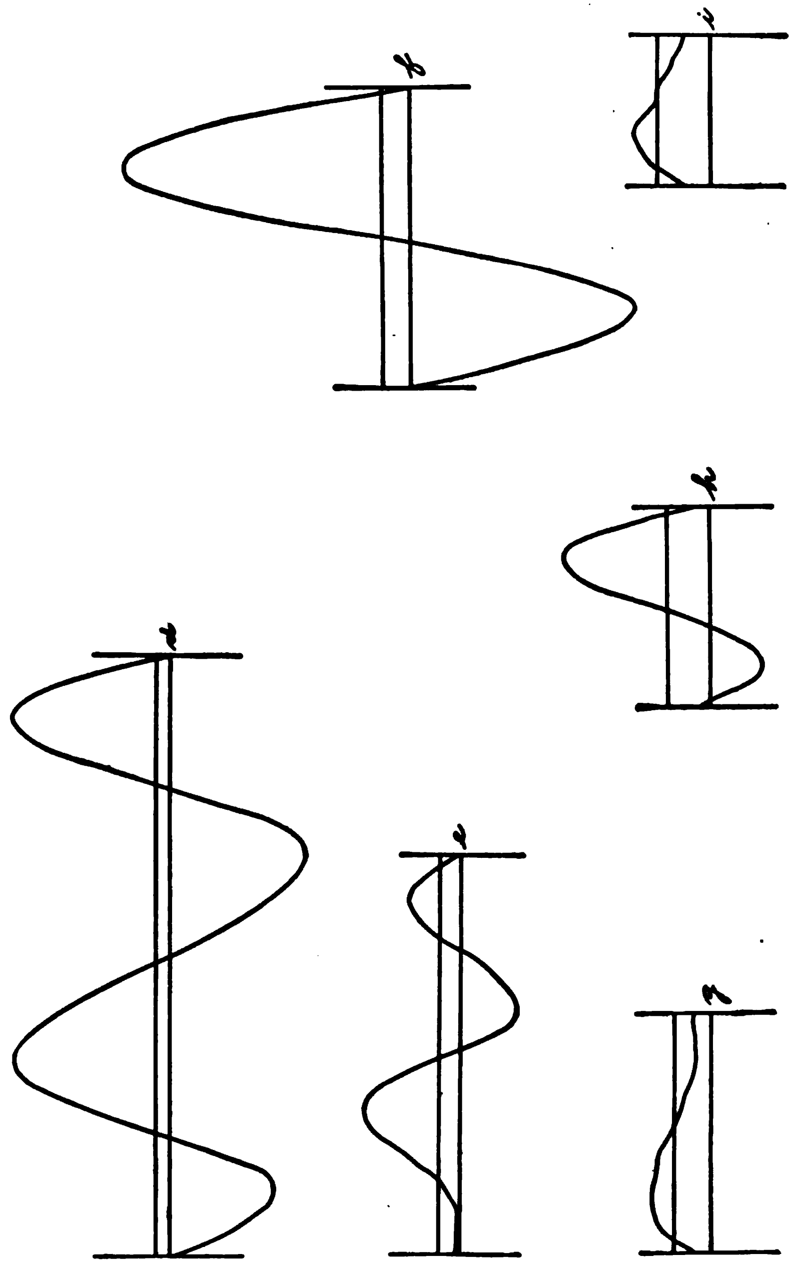


Fig. 129 (d to i).

$g =$ " " five- " " $\frac{720^\circ}{5} = 144^\circ$;

$h =$ " " six- " " $\frac{720^\circ}{6} = 120^\circ$;

$i =$ " " eight- " " $\frac{720^\circ}{8} = 90^\circ$.

The data from these diagrams other than reversals which can be read off at a glance are tabulated below.

NUMBER OF CYLINDERS.

	1		2		3		4		5		6		8	
Initial inertia....	26	208	26	208	26	208	26	208	26	208	26	208	26	208
P..............	48.8	.....	....	.....	....	....	....	.....	....	....	....	.....	....	48.8
F_t.............	7	7	14	14	21	21	28	28	35	35	42	42	56	56
$\frac{\text{Max. turning force}}{\text{Mean turning force}}$	7.82	14.57	4.57	13.50	3.42	5.50	1.68	17.59	2.14	2.06	1.38	3.80	1.44	1.66
$\frac{\Delta E}{E}$.............	.78	.84	.73	1.56	.57	.59	.22	2.26	.29	.19	.15	.59	.13	.11

Extremely significant is the manner in which these ratios change with inertia and cylinder combination after what has been said on balance. The results will be referred to again and used in designing crank-shafts and fly-wheels.

PART III.

DIMENSIONS OF THE ENGINE PARTS.

67. Cylinders.—From what has been presented in Part I., the diameter stroke and R.P.M. for a single cylinder of the required power can be calculated and the probable economy predicted. Considerations of balance or turning effort may show some particular multiplication of cylinders with some special grouping to be desirable. In this case the same displacement per minute will give the required output when divided among the desired number of cylinders. Having the number of cylinders, their general arrangement, the stroke and diameter, there remains the filling-in of details of requisite form and dimensions to secure the result in physical actions and to resist the stresses set up by gas pressure in the cylinder and the kinetic forces of such moving parts as are needed.

68. Cylinder Walls.—A cylinder resisting an internal pressure is subject to simple tension if not too thick. The maximum gas pressure that may ever be met is the pressure that the metal must be able to resist. In gas-engines of whatever compression or fuel this probable maximum pressure may be calculated by the pressure ratios applied to the pressures of compression as explained. The highest possible, though normal, pressure may be taken as 450 lbs. per sq. in. for engines with high compression and rich fuel.

Occasionally an explosive wave may develop, especially when the space containing the charge is very irregular and consists of isolated pockets. This wave may generate momentary pressures at the crest as high as 600 lbs. per sq. in., but if the maximum of 450 lbs. per sq. in. be taken with a sufficiently high factor of safety or sufficiently low initial material stress the results will be perfectly satisfactory. The probable maximum is to be calculated as explained in Part I under pressure ratios.

Consider a cylinder of unit length 1 in.

There will act to disrupt along an element parallel to the axis a force F pounds.

Let P_m = maximum pressure pounds per square inch;

d = diameter of cylinder in inches;

t = thickness of cylinder wall in inches;

S = resisting stress in pounds square inch of metal in tension;

then

$$F=P_m\times d \text{ pounds.}$$

This is resisted by two thicknesses of metal, one at each end of a diameter; hence the resisting force is

$$R=2t\times S.$$

If these are to be equal,

$$P_m\times d=2t\times S,$$

or

$$t=\frac{P_m\times d}{2S}. \quad \ldots\ldots\ldots \quad (75)$$

This form of the expression, then, for cylinder thickness and its value depends on the selection of S.

Cylinders after use require reboring, and a certain amount is allowed for this; hence to the above should be added a constant c (sometimes a function of d), giving

$$t=\frac{P_m\times d}{2S}+c, \quad \ldots\ldots\ldots \quad (76)$$

to resist the bursting along a longitudinal element.

The gas pressure acts on the head and frame, also tending to part the cylinder along a circle. For this case

$$F=P_m\times\frac{\pi d^2}{4}.$$

The total metal resisting is a ring of internal diameter d and external diameter $d+2t$; hence the area of the ring is

$$\frac{\pi}{4}[(d+2t)^2-d^2]$$
$$=\frac{\pi}{4}[d^2+4dt+4t^2-d^2]$$
$$=\pi t[d+t].$$

Hence the resistance by the metal to this stress will be

$$R=\pi t[d+t]S.$$

Equating this to the force acting,

$$\pi t[d+t]S=\frac{P_m\times\pi d^2}{4};$$
$$\therefore\ t^2+td=\frac{P_m d^2}{4S}.$$
$$t^2+td+\left(\frac{d^2}{4}\right)=\frac{P_m d^2}{4S}+\frac{d^2}{4};$$
$$\therefore\ t=\frac{d}{2}\left(\sqrt{\frac{P_m}{S}+1}-1\right). \quad . \quad . \quad . \quad . \quad (77)$$

The thickness must be great enough to resist both these stresses and if one be greater than the other the greater will include the less. To find which is greater assume

$$P_m=399,$$
$$S=1;$$

then

$$t=\frac{d}{2}\times 19=9.5d,$$

or

$$t=\frac{399d}{2}+c=.199.5d+c.$$

Hence t by formula (76) will be larger, as c is always small compared with d.

If therefore the thickness of cylinder be taken as

$$t=\frac{P_m d}{2S}+c=225\frac{d}{S}+c,$$

or

$$t=\frac{P_m d}{2S}+c'd=d\left(\frac{225}{S}+c'\right),$$

and S properly selected, the result will be satisfactory for strength, but may be too heavy, for here $P_m=450$ lbs. per sq. in.

69. Empirical Formulas for Cylinder Thickness.—For particularly sound castings high values of S may be used. This is otherwise expressed by making t increase not as the diameter directly but as some fractional power of d with a constant S. The problem is very similar to that for steam-cylinders, but the many formulas for steam-cylinders give thicknesses that are different.

Thurston gives $$t'=aP_m d+b. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (78)$$

$a=.0004$ for short strokes or vertical cylinders
to $a=.0005$ for long strokes or horizontal cylinders;
$b=.5$ for horizontal cylinders
to $b=.2$ for vertical cylinders.

Hence in this case

$S=1250$ lbs. per sq. in.
to $S=1000$ " " " "

while the reboring constant varies with the tendency to wear.

Seaton gives $S=1250$ for cast iron
to $S=2500$ " " "

and for large marine engines the reboring constant

$$c=0.6''.$$

For large cylinders with low pressures Weisbach gives

$$t = .00033 P_m d + 0.8; \quad \ldots\ldots \quad (79)$$

in which

$$S = 1500,$$
$$c = 0.8.$$

Van Buren gives

$$t = .0001 P_m d + .15\sqrt{d}, \quad \ldots\ldots \quad (80)$$

in which

$$S = 5000,$$
$$c = .15\sqrt{d},$$

the reboring thickness proportional to the square root of the diameter. He also gives

$$t = .03\sqrt{P_m d} = \frac{P_m d}{33\sqrt{P_m d}}, \quad \ldots\ldots \quad (81)$$

in which the reboring term is omitted, or, rather, included.

Here $S = 16.5\sqrt{P_m d}$, making the stress allowable proportional to the $\frac{1}{2}$ power of both maximum pressure and cylinder diameter. This covers rigidity and gives usually large thicknesses.

Haswell's formula is the same in form as Weisbach's, but he uses

$$c = .125'',$$
$$S = 1250.$$

For small cylinders Whitham makes $c = 0$ and $S = 165\sqrt{P_m}$, which is similar to Marks's form, where

$$c = 0,$$
$$S = 1800.$$

Using many formulæ, Kent calculates the diameter of cylinders of different sizes and finds a formula representing a mean of the results He gives

$$t = .0004 P_m d + 0.3'',$$

in which

$$c = 0.3,$$
$$S = 1250 \text{ lbs. per sq. in.}$$

70. Empirical Formulas for Gas-engine Cylinder Walls.—For gas-engine cylinders H. H. Kelly gives $t = .095d$ for cylinders 7″ to 15″ in diameter. Taking 450 lbs. as P_m this involves

$$S = \frac{450}{2 \times .095} = 2367 \text{ lbs. per sq. in.}$$

For small gas-engine cylinders $d = 5''$ to $8''$, Güldner gives, when cast with the frame,

$$\begin{aligned} & S = 1500 \text{ lbs. per sq. in.,} \\ & c = .13'' \\ \text{to} \quad & c = .20. \end{aligned}$$

When larger and separate, higher values may be taken for S and larger allowance for reboring.

Mr. S. A. Moss gives, on comparing reports of dimensions used in American practice,

$$\begin{aligned} & c = \tfrac{1}{4}'', \\ & S = 1625 \\ \text{to} \quad & S = 3750 \text{ for } P_m = 300 \text{ lbs.} \end{aligned}$$

This is too low a value for P_m, for, except in a continually throttled engine, the maximum *possible* pressure will exceed this. Taking a maximum of 450 lbs. the metal stress for the above case is

$$\begin{aligned} & S = 2437 \\ \text{to} \quad & S = 5625. \end{aligned}$$

While some engines are running satisfactorily with this high stress, it is somewhat too high for good practice. Strange to say, too, the higher stress is not always confined to large engines, as it should be.

For small automobile engines where lightness is a consideration reboring provisions may be omitted, and

$t = \frac{1}{4}''$ for cylinders $1\frac{1}{2}''$ up to $2\frac{1}{2}''$,
$t = \frac{3}{8}''$ " " $2\frac{1}{2}''$ " " $5''$.

For cylinders abo~~ut~~ve 5″ the formula should be used. No cylinder should be less than ¼″ thick. In these cylinders a maximum of 300 lbs. is rarely exceeded, as they all use vacuum carburetors. This corresponds to a stress for the four limiting cases

$$S = 900 \text{ to } 1500$$

and

$$S = 1000 \text{ to } 2000.$$

A formula allowing 3000 lbs. stress is $t = .075d$, and many engines are built according to this.

For medium-sized engines of diameters 6 to 15 ins. reboring must be allowed for and the following may be used:

$$t = .060d + .2, \qquad \dots\dots \qquad (82)$$

for ordinary work;

$$t = .075d + .3, \qquad \dots\dots \qquad (83)$$

for heavy work and stiff cylinders.

This corresponds, on a basis of 450 lbs. maximum, to stresses

$S = 3750$ lbs. per sq. in.,
$S = 3000$ " " " "

For large cylinders, if needed very stiff to reduce vibration period of frame,

$$t = .085d + .3, \qquad \dots\dots \qquad (84)$$

on a stress of 2648 with 450 lbs. maximum. This is greater than needed for strength.

Large two-cycle cylinders must be built heavier than four-cycle to resist the weakening in strength and stiffness due to the exhaust ports. For this

$$t = .12d + .3, \qquad \dots\dots \qquad (85)$$

involving 1870 lbs. stress on 450 lbs. maximum gas pressure. There is always danger of overheating these thick cylinders, so that the greatest caution should be observed in applying excess metal.

For regular work where no unusual stiffness is required

$$t = .052d + .3 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (86)$$

will serve.

This involves a stress of $S = 4327$ on a basis of 450 maximum gas pressure.

The entire cylinder length need not be equally heavy, as pressures fall rapidly with the piston motion; hence, in large cylinders especially, the walls may taper somewhat.

Güldner gives

$S = 4300$ lbs. per sq. in. allowable,
$c = .25''$.

He says "excessive thickness of walls is objectionable on account of difficulty in getting dense castings and difficulty of cooling. When cylinder diameter exceeds 2 ft. the thickness may decrease till the factor of diameter is only two-thirds that for part receiving maximum pressure."

71. Cylinders too Thick for Simple Tension in Walls.—The preceding work supposes the thickness of the cylinder so small in comparison with the radius that the whole cross-section is equally in tension. Very high pressures, with the necessarily very thick walls, involve consideration of three facts:

(*a*) A difference in stress in inner and outer cylindrical layers, it being greatest on interior layers.

(*b*) The elongation of interior layers to permit outer layers to receive tension results in decreased area.

(*c*) This decreased area of cross-section from inside stretch increases the bore.

More exact formulas take into account one or more of these actions.

The simple formula for a thin shell is

$$\frac{t}{r} = \frac{P_m}{S}.$$

Brixes' formula neglects (*c*) and is given by

$$\frac{t}{r} = (l)^{\frac{P_m}{S}} - 1 = (2.718)^{\frac{P_m}{S}} - 1, \quad . \quad . \quad . \quad . \quad . \quad . \quad (87)$$

where *e* is the logarithmic base 2.718.

Barlow's form is exact and covers (*a*), (*b*), and (*c*) by

$$\frac{t}{r}=\sqrt{\frac{S+P_m}{S-P_m}}-1. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (88)$$

The effect of making the internal pressure approach the tensile resisting stress in the metal is pointed out by Mr. Alfred Petterson in the "American Machinist." These results are given in the table. He takes allowable tensile strength of 1800 lbs. per sq. in. for cast iron.

VALUES OF $\frac{t}{r}$.

$\frac{S}{P_m}$.	Simple Thin Sheet, $\frac{t}{r}=\frac{P_m}{S}$.	Brixes, $\frac{t}{r}=(l)^{\frac{P_m}{S}}-1$.	Barlow, $\frac{t}{r}=\frac{P_m}{S-P_m}$.	Lamé, $\frac{t}{r}=\sqrt{\frac{S+P_m}{S-P_m}}-1$.
1.667	....	....	....	1.0000
2.000	.500	.6487	1.0000	.7320
3.000	.333	.3956	.5000	.4142
4.000	.250	.2840	.3333	.2910
5.000	.200	.2214	.2500	.2247
6.000	.167	.1813	.2000	.1832
8.000	.125	.1331	.1428	.1339
10.000	.100	.1051	.1111	.1055
12.000	.083	.0869	.0909	.0871
15.000	.067	.0689	.0714	.0690
20.000	.050	.0512	.0526	.0513

As the highest pressure ever reported in gas-engine cylinders, even for the explosive waves, is 600 lbs. per sq. in., with an allowable tension of 6000 lbs. per sq. in. the ratio is $\frac{S}{P_m}=10$. For this value of the ratio $\frac{S}{P_m}$ the table shows the ratio $\frac{t}{r}$ will not differ from the value for simple tension more than .011 by Barlow's or .0055 by the exact Lamé formula. It will appear from this that the form of formula used is justifiable.

73. Cylinder-heads and Breech Ends.—Small cylinders are cast with the head in one piece with the walls and jackets. Openings are left for valve insertion, and usually also in the jacket-head to facilitate coring. Such openings are closed by ground-joint flat plates held by a single bolt at the center or by yokes of cast iron, one yoke and one bolt holding two covers. These small cylinder-heads are usually hemispherical, in which case

$$\text{Head thickness } t' = \tfrac{1}{2}t \text{ for strength;}$$

but as these walls are seldom designed for strength, but rather for soundness of casting, such heads are to be made of the same thickness as walls. Even when hemispherical or approximately so the valve-chamber forming extensions of the clearance space will have flat sides and all openings for valve removal will be covered by flat circular plates. All such flat sides and plates are subject to the uniformly distributed internal pressure, which may be calculated by compression-pressure ratios for these engines or assumed to be 300 to 450 lbs. per sq. in. for the highest possible. These plates when forming part of the casting are flat plates secured along the edges.

When held by a bolt through the center, or by a yoke and bolt pressing on the center, they are supported in the middle apparently, but since the ground joint to be tight must be under compression greater than the internal gas pressure initially they must rather be considered as flat circular plates supported at the edges and loaded in the middle. The load is, moreover, uncertain and depends on the tension of a long bolt passing through the yoke and secured to the cylinder-head. Under these circumstances it is clear that the actual load these flat plates will be caused to support, and the stress in the metal, cannot be computed. A more empirical rule, then, will be given for these plates, covering a hole whose diameter is

$$\text{Cover-plate hole diameter} = \text{outside valve diameter} + \tfrac{3}{16} \text{ inch.}$$

The other dimensions of the plate are usually a little heavier than the valve, which is designed as an edge-supported plate with extra metal for stiffness.

Cover-plate yokes are beams loaded in the middle and supported at both ends, and their strength will depend on cross-section, distribution of metal, and the conditions of application, and their form will vary so much that no general formula can be given for them. They must carry the maximum gas pressure on one or two cover-plates in addition to whatever stress is necessary for tightness, and the whole is to be resisted by a single bolt.

No yoke-bolt should be less than ½″ diameter or it may be too easily stripped; ⅝″ would be better. Too large a bolt or too short a one would not give the necessary elongation to put proper stress on the cover-plates without breaking. In short, such bolts on screwing up nut will increase load to breaking strength of bolts too quickly. Too long a bolt makes shearing easy. Practically such bolts are to be ⅝″ or ½″ diameter, sometimes ⅜″ on small engines, and never used on engines of greater cylinder diameter than 6″, nor should they have less than 1½″ of clear bolt in the yoke.

The cylinder casting should be so thickened where the yoke stud enters as to give a thread support at least equal to that offered by the nut, and the surrounding walls should be stiffened to resist the load resulting from the pull on the casting.

74. Volume of Clearance in Breech End.—The clearance space is usually so irregular in form in small engines as to be difficult to evaluate volumetrically. If, therefore, the volume be calculated as correctly as possible, and the dimensions so selected as to give the ratio by the formula below, the compression pressure desired can be obtained by adjustment after the first engine is built.

Let clearance volume $=v_b=$ volume of gases at end of compression in cubic inches;
" compression pressure in atmospheres $=p_b'$;
" compression pressure in pounds per square inch absolute $=p_b$;
" atmospheric pressure in pounds per square inch absolute $=14.7$;
" displacement in cubic inches $=\delta$;
" total volume before compression $=v_b+\delta$ cubic inches;
" cylinder diameter $=d$ inches;
" crank radius $=\frac{l}{2}$ inches;

then, very nearly,

$$v_b=(v_b+\delta)\left(\frac{14.7}{p_b}\right)^{\frac{3}{4}}=(v_b+\delta)\left(\frac{1}{p_b'}\right)^{\frac{3}{4}}. \quad . \quad . \quad . \quad (89)$$

Hence

$$v_b\left[1-\left(\frac{1}{p_b'}\right)^{\frac{3}{4}}\right]=\frac{\delta}{(p_b')^{\frac{3}{4}}}$$

and

$$v_b = \frac{\delta}{(p_b')^{\frac{3}{4}} - 1} = \frac{\pi d^2 l}{4} \times \frac{1}{(p_b')^{\frac{3}{4}} - 1} = \frac{.785 d^2 l}{(p_b')^{\frac{3}{4}} - 1} \quad . \quad . \quad . \quad (90)$$

This is the clearance volume in cubic inches to give the compression of p_b' atmospheres, very nearly.

75. **Adjustment of Clearance.**—This quantity v_b can be computed easily from the tables of cubes and fourth roots. Adjustment of clearance volume or compression pressure after contraction can be made by

(*a*) Changing the connecting-rod length.

(*b*) Arching or depressing cylinder-head or clearance walls.

(*c*) Arching or depressing piston-head.

(*d*) Attaching plates or other masses to the interior walls of cylinder or piston-head.

(*e*) Cutting of metal from interior parts originally left a little thick for this purpose.

The attachment of any mass to the interior of the clearance to increase compression will undoubtedly result in preignitions from overheating unless it is possible to get such thermal contact as to carry off heat as fast as absorbed. This is impossible with very thick metal.

The best method is either that of arching or cutting. To do the latter, the piston-head can be made quite thick and clearance calculated smaller than desired. This may leave the piston heavier than is desired, but this is not necessarily bad, as reference to crank efforts will show. Arching or depression of walls requires slight alteration of pattern and certainly new castings. Connecting-rod adjustment requires first a true cylinder bore beyond where the piston is intended to stop and in some connecting-rods can be secured by simply using liners or shims behind the brasses.

(*f*) Another adjustment not as simple to practice but very effective consists in lengthening or shortening the frame between main bearings and cylinder. If the frame and gear permit, this is a good method, but very few valve-gears are so constructed as to make this possible. A two-part piston would make adjustment easy by a very effective method.

In some engines even of considerable size the valves are located directly in the middle of a flat head; when this is the case, the clearance is simply a cylindrical extension of the cylinder of the same bore. Its cross-section is the same as the cylinder and its length λ is given by

$$\lambda = \frac{l}{\left(\frac{1}{p_b'}\right)^{\frac{1}{n}} - 1}. \quad \quad (91)$$

76. Flat Heads.—In cases like this the flat head with the valves constitutes a circular plate uniformly loaded, except where the valves enter, and loaded around circular lines where valves pass through. The head is jacketed and the various webs connecting jacket and head proper that make up the valve ports and supports make of this double-walled head a circular girder in effect. While, therefore, the formula for a circular plate uniformly loaded may be applied it is useful as a guide only, since

1°. There is not a single plate, but two plates connected by webs.

2°. The bottom plate is not uniformly loaded.

3°. It is not uniformly supported at the edges, as it is held by bolts.

4°. It is not merely supported, but partly supported and partly fixed by the compressive effect of the bolts and broad face of contact which extends across cylinder walls, jacket space, and jacket walls, the bolts usually entering lugs in the jacket space.

The formula of Grashof for uniformly loaded plates is

$$T = r'\sqrt{\frac{2}{3}\frac{P_m}{f}} \quad \quad (92)$$

when fixed at edge;

$$T = r'\sqrt{\frac{5}{6}\frac{P_m}{f}} \quad \quad (93)$$

when supported at edge,

where r' = active radius in inches;

T = thickness in inches;

P_m = maximum pressure pounds per square inch;

f = working stress in the metal.

Comparing this with formula for cylinder thickness and putting $f=S$ and P_m the same,

$$t=\frac{P_m r}{S} \quad \text{or} \quad P_m=\frac{St}{r};$$

substituting and making $r'=r$,

$$T=r\sqrt{\frac{2}{3}\frac{St}{rf}}=\sqrt{\frac{2}{3}rt} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (94)$$

for fixed edge; plate thickness relative to cylinder thickness with equal working stresses. Thickness of heads proper, when web-attached to jackets, is never calculated from the plate formulas for reasons given, but is given as a function of the first powers of t. This is also done in steam work.

Flat part of heads will then have thickness:

$$T'=0.7t \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (95)$$

$$\text{to} \quad T'=1.4t, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (96)$$

depending on how much flat surface is exposed and how much truss effect is secured by jacket-webs. For small surfaces and good connection use the lower figure, for large flat surfaces use the larger. For steam-engine flat heads Kent gives as the average of many formulas

$$T''=.00036dP_m+.33. \quad . \quad . \quad . \quad . \quad . \quad . \quad (97)$$

77. Valve Location in Heads.—Placing valves in the head to get simple clearances involves the use of valve-gear on the head itself, and though by removing the head all valves are exposed, this advantage is nil in large engines when the part is heavy and work difficult. For this reason the majority of gas-engines distribute the valves differently. Valve-stems should be vertical, else the seating will be eccentric. The location of valves with vertical stems in positions that will make them easily removable and yet not strike the piston makes it necessary to introduce them through offset spaces in the clearance. These valve-pockets and the passages connecting them to the cylinder proper will constitute an irregular casting called the breech end.

Valve-pockets may be offset from the cylinder itself wholly; from the head or breech end proper wholly or partly from both; the dividing line between breech end and cylinder is the joint.

In small vertical cylinders the inlet will usually be carried in the head with its stem projecting upward and immediately below it the exhaust-valve will seat on the cylinder casting, the stem projecting downward.

In this case there will be two offsets, the one in the head matching the one in the cylinder. When this is done the inlet-valve is usually housed in a cage, removal of which permits removal of exhaust-valve upward from its seat in the main casting. Both stems vertical and both projecting downward, in vertical engines is secured by putting both on same side with one wide offset to carry them or one each on opposite sides of the cylinder with two opposing offset pockets. When horizontal cylinders are used a single offset on the end may carry both valves either one above the other and stems one up and one down, or both pointing the same way. The former requires a narrow offset and the latter a wide valve-pocket offset. The same result can be secured by putting both on one side or one on each side or one on the end and one on the side. No matter what arrangement is used to make both valves easily removable and work on vertical stems irregular breech ends are necessary, and the cylinders may be irregular to match or, as in case of both valves in the end, the cylinder may be perfectly plain.

78. Breech-end Walls.—The thickness of the walls of those irregular passages can always be more or less correctly judged from the knowledge of the necessary thickness of the true cylindrical surface and the thickness of a flat plate. This, however, is based on the assumption that the resistance of the metal is known or knowable. With the irregular castings resulting from involved breech ends with their jackets connecting webs, gas, air, and exhaust ports, bolt-holes for connection to cylinder, lugs, etc., for connecting valve-gear, there are many unknown and perhaps serious cooling strains in the metal. The best results will be secured by as great simplicity as possible and by dividing into parts to be bolted up. Wherever possible these breech ends, irregular at best, should be cast separate from the cylinder, which so far as possible should be a plain shell to insure truth for varying temperatures.

79. **Head and Breech-end Fastening.**—Breech ends when separately cast are secured to the cylinder by bolts. Some builders have used long bolts extending the length of the cylinder, securing breech to cylinder and cylinder to frame with one fastening. This is not good for two reasons: first, there is a different coefficient of expansion for wrought-iron bolts and cast-iron cylinder; secondly, to secure the compression at the joint necessary for tightness some elongation must be put on the bolts, and after setting up for tightness when a little warm, cooling may result in permanent set and cause, first, leakage and later failure. Long bolts also are subject to excessive shear in setting up more than is desirable. Cylinder-heads and breech-end bolts are not as a rule secured through flanges as on steam-engines, but through solid metal in the jacket space. Stud-bolts are most common, and the cross-section necessary for strength is easily calculated:

Let A_c = cross-section of clearance space along the joint line, the maximum load is

$$P' = P_m A_c \text{ pounds.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (98)$$

The cross-section of all bolts combined must be such as will not call for more than 4500 lbs. per sq. in. fiber stress for bolts less than $\frac{7}{8}''$ and upward for large bolts to 7000 lbs. Reference in this connection should be made to Cathcart's "Machine Design," Vol. I. The number of bolts and their size will be dependent not on strength but on the conditions for securing a good joint. To make a joint between breech end and cylinder, either a thin asbestos gasket is used or a finely machined scraped or ground joint is made. When water-jacket circulation is made through the joint the gasket is used, but with the fine joint an outside connection either by bolted-up castings or by piping must be made between breech and cylinder-jacket or a separate circulation system used; this last is best for large engines.

With an asbestos gasket joint considerable compression must be put on the gasket to prevent leaks, especially between bolts. Hence the bolts should be as close together as possible, allowing only proper freedom for the wrench. They should not be too large in diameter or a man must apply a hammer to the wrench in setting up, which results, perhaps, in too much stress. If too small, they are subject to twisting off, and so must be large enough to prevent an ordinary man from applying stress enough on the wrench to break. The proper location of bolts will depend on the form of the joint, which is seldom circular, special care being exercised at corners.

Professor Cathcart, quoting Professor Barr, says that the initial load P_1, due to the mechanic's handling of the wrench, may be estimated at about 16,000 lbs. per in. of bolt diameter. The intensity of the initial tensile stress in bolt varies inversely as the diameter and will be about

$$f_b = \frac{30000}{\text{bolt diameter}} \text{ lbs. per sq. in.} \quad . \quad . \quad . \quad . \qquad (99)$$

In addition to this tensile stress there is considerable twisting of the bolt.

The following data on wrenches represent approximately the practice of J. H. Williams & Co. in making wrenches by drop-forging.

Let d_b = diameter of bolt in inches;

then

Opening between jaws $= 1.5\,d_b + \frac{1}{8}''$; . . . . (100)

Width of metal beyond " $= d_b + \frac{1}{32}$ for small sizes; . (101)

" " " " " $= d_b + \frac{1}{8}''$ " large " . . (102)

Hence for small wrenches the spacing of bolts should be:

Opening between jaws $= 1.5d_b + \frac{1}{8}''$

+width of metal around jaw $= d_b + d''$,

where $d'' = \frac{1}{32}''$ to $\frac{1}{8}''$

+a distance from wrench to next unit $= d'$;

giving

Distance between bolt centers $= 2d_b + \frac{1}{8}'' + d' + d''$. . . (103)

The clearance d'' may be made as small as in the designer's judgment seems desirable.

The center line of the stud-bolts should be such that the bolt comes tangent to the cylinder walls on the outer side. Usually there will be sufficient metal outside the bolt for threading the casting to receive the bolt thread; if not, the jacket must be thickened to give at least a bolt diameter of metal outside and all around.

What fastenings or connections will be needed on the cylinder and breech cannot be determined until the valve, governor, and igniting gear have been laid down, and then the stresses may be analyzed and strength checked, but rigidity is most important.

80. Cylinder Length.—The length of the cylinder will depend on the length of the piston, the stroke, and the form of the breech end in connection with the compression pressure. After laying down the breech end the inner point of the piston travel is determined; from this the length of piston can be laid down and the length of cylinder to always cover the piston will be beyond this one stroke length. Pistons generally project beyond their cylinders when single-acting, especially when short frames are desired.

81. Water-jackets.—Every part in contact with the hot gases should be water-cooled except in very small engines. As the piston-head never reaches the crank end of the cylinder the jacket seldom extends the length of the cylinder; the usual practice is to extend the jacket a short distance beyond the point where the piston-head stops on the outstroke. All the breech-end parts must have water contact, and when large, the piston- and exhaust-valves must also be jacketed or their centers will overheat. Igniters are sufficiently cooled by making a good, machined contact between the supporting barrel and some water-cooled part.

The amount of water that is necessary should be such as will carry off 50% of the heat generated by the combustion. Frequently as low as 30% will be thus carried off with early ignition and long expansion, but 50% should be the amount of heat to be counted on. Another way of computing is to provide for the carrying off of twice as much heat by the jacket as is converted into the work. Thus a 50-I.H.P. engine should be provided with water enough to carry off

$$\frac{50 \times 2 \times 33000}{778} \text{ B.T.U. per minute.}$$

This is only approximately true, but errs on the right side.

Let w = pounds water per hour required;
$t-t'$ = temperature rise allowed, degrees F.;
E_3 = thermal efficiency based on B.T.U. in fuel (high) and I.H.P.;
Q' = B.T.U. per hour set free by combustion;
Q'' = " " " to be carried off by jacket;

then heat per hour set free in engine is

$$Q' = \frac{\text{I.H.P.}}{E_3} \times \frac{33000}{778} \times 60 \text{ B.T.U. per hour.} \quad . \quad . \quad (104)$$

Heat to be carried off is

$$Q'' = .30Q'. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (105)$$

$$\text{to} \quad Q'' = .50Q'. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (106)$$

Heat absorbed per pound water is

$$t-t' \text{ B.T.U.};$$

hence the pounds water per hour required is

$$\text{from} \quad w = \frac{.30}{t-t'} \frac{\text{I.H.P.}}{E_3} \times \frac{33000}{778} \times 60 \quad . \quad . \quad . \quad (107)$$

$$\text{to} \quad w = \frac{.50}{t-t'} \frac{\text{I.H.P.}}{E_3} \times \frac{33000}{778} \times 60. \quad . \quad . \quad . \quad (108)$$

The maximum temperature t may be about 180° F.; it may be higher if care has been exercised in preventing the formation of steam pockets in which the generation of steam may prevent water entering.

Should the water be allowed to boil and the jacket be constructed for this a much smaller amount of water will be needed. Taking 960 B.T.U. as the latent heat of steam at the pressure in the jacket slightly above atmosphere the water necessary will be

$$w' = \frac{.30}{960} \frac{\text{I.H.P.}}{E} \times \frac{33000}{778} \times 60 \quad . \quad . \quad . \quad . \quad (109)$$

$$\text{to} \quad w' = \frac{.30}{960}\frac{\text{I.H.P.}}{E}\times\frac{33000}{778}\times 60. \quad \ldots \quad (110)$$

Small stationary engines sometimes use convection circulation between the cylinder and an elevated tank. In such cases the rate of cooling must equal that of reception of heat or loss of weight will occur by boiling. In fact boiling does generally occur in such installations if the load be heavy and run long.

Automobiles use pumps for circulation and the capacity of the pump is given above by w. The water is then cooled by a radiator which must be able to extract w B.T.U. per hour; if not the water will boil here also. If the radiation were sufficiently effective heat might be taken from the cylinder walls directly, and in some automobile engines by inserting threaded rods like quills the proper heat absorption is secured if a strong air-blast is available.

The water-cooled automobile engine simply uses water to increase the effective air-cooling or radiating surface, in which case a circulating pump is necessary.

The centrifugal circulating pump of properly small size when tested by M. Butin gave efficiencies from 18 to 35% on pressures ranging from 8 to 15 ft. of water. The rate at which the water is air-cooled depends on the difference in temperature of air and water, the air velocity, the water velocity, and the nature and extent of the surface. Fans are generally used to increase air velocity; increase of surface is secured by many small tubes, metal discs on pipes, etc. MM. Grouville and Arquembourg report the surface per horse-power for the disc and pipe radiator, the section normal to the draught, as

34 square inches when one tube deep,
18 " " " two tubes "
13 " " " three " "

So that for the best case there is abstracted per square inch of projected surface per hour about

$$\frac{.50}{34\times E}\times\frac{33000}{778}\times 60 \text{ B.T.U.},$$

or taking $E_3 = .125$, the radiation rate is about 300 B.T.U. per hour per square inch of air-cooled surface.

82. Water Space.—The water space must be so formed as to permit true circulation without eddy currents and without steam- or air-pockets.

Water heats in the jacket and convection currents are set up which are greater the slower the flow through the jacket and the larger the space. No possibility of mud settling in places without means for removal should be permitted. The jacket may be cast in one piece with the cylinder, but better castings will result, especially in large work, if it be separately cast and fitted. This, however, is more expensive.

When separate jackets are used the water space is generally larger than when cast; to the thin shell there are added internal rib webs for stiffness.

Let w_1 = width of water space;
then, for usual practice in horizontal cylinders,

$$w_1 = .06d. \qquad (111)$$

For vertical cylinders with good circulation the space is less than this, especially when several separate feeds enter jacket divided longitudinally, making several individual jackets. More heat will be carried away in a given time by a swift thin stream than by a slow thick one and the surfaces kept cleaner. For this make

$$w_2 = .045d. \qquad (112)$$

When large cylinders have separate jackets w may be quite large.

$$w_3 = .15d. \qquad (113)$$

If for any structural reason it is desired to alter the water space, it may be done without harm provided in so doing no dead ends or pockets or misdirected circulations result.

83. Jacket Walls.—A jacket wall must resist internal water pressure if there be any, and then it seldom exceeds the 50 lbs. per sq. in. in the city main. The jacket wall must also, which is more important, carry parts of the mechanism and often the whole weight of the cylinder when horizontal. In cases where it has extra service to perform, especially where stiffness is required, it must be considered a part of the frame and be properly webbed.

Small automobile engines have jacket-wall thickness,

$$j = .06d. \qquad (114)$$

Slow-speed stationary engines, where stiffness is more important than lightness,

$$j_1 = .10d. \qquad (115)$$

With proper reinforcements at the necessary places

$$j_2 = .05d. \qquad (116)$$

This latter is most common.

Where the separately webbed jacket is used

$$j_3 = .025d. \qquad (117)$$

Breech ends, when irregular, usually have water space irregular in thickness to give the outside a uniform appearance. Wherever by reason of the layout the jacket must support or transmit stress, the thickness must be made great enough to resist this stress, and analysis of such maximum stresses is therefore necessary for each specific case.

To permit of a cold-water-pressure test of 400 lbs. per sq. in. for leakage Güldner recommends a thickness about

$$j_4 = .022d, \qquad (118)$$

provided the stress of about 8500 lbs. per sq. in. is not often repeated.

83. Cylinder Charges and Valves.—Entrance and exit of gases through the valves should be accomplished with as little expenditure of energy as possible, so that during entrance a full charge shall be secured and during exhaust expulsion be complete and uniform.

To minimize the energy necessary for the motion of gases, suction should take place at pressures as little below atmosphere as possible and exhaust as little above atmosphere as possible. These pressure differences through the valves will depend on the velocity of the gases, which in turn depends on the piston speed and valve opening. Valve openings, then, should depend on the piston speed, and as this is variable during the stroke, while the valve diameter is constant, it is evident that the lift of the valve should vary with piston speed to keep gas velocity uniform.

Vacuum on the suction stroke may be caused by the demands of a carburetor in gasolene-engines; hence in these cases the vacuum in the cylinder is doubly low, one part being due to the carburetor and the other to the valve opening, with possibly some due to passages that are too small or crooked.

With high piston speeds and short strokes—that is, a high R.P.M.—the mass of mixture must be started in motion toward the cylinder at a velocity of several thousand feet per minute. This motion must be arrested and the valve closed in a very short space of time.

At 1800 R.P.M. a stroke is completed in 1/3600 minute, or 60/3600 = .0166 second.

This will bring into play inertia of the gases; hence loss on charging may be due to this in cases of high alterations per minute, and to reduce the loss the valve should be held open longer than the end of stroke.

Back pressure may result from too small an opening in mufflers or improper setting. This is more serious the higher the terminal pressure, and hence worse with low compressions, rich fuels, late ignition, and high exhaust velocity, especially when speeds are high at the same time because of inertia effects in the gases.

84. Gas Velocity through Valves and Piston Speed. — Gas velocities have been usually computed on the basis of full valve lift for the stroke in a somewhat similar way as piston speeds are computed on an assumed constant velocity for the stroke. There are no serious consequences arising from the latter, but with the former, unless the cam is so formed that the lift is at all times proportional to actual piston speed, there may be a serious loss at certain parts of stroke.

The real piston velocity is given approximately by the following formula, which neglects terms higher than 2θ.

$$S' = v(\sin\theta + \tfrac{1}{2}a_1 \sin 2\theta), \quad \ldots\ldots \quad (119)$$

where $a_1 = \frac{r}{l}$;

θ = crank-angle;

v = linear velocity of crank-pin, feet per second;

S' = velocity of piston, feet per second.

The following table gives the values of the parenthesis for crank-angles during the outstroke.

θ.	$(\text{Sin}\ \theta + \frac{1}{2}a_1 \sin 2\theta)$.			
	$a_1 = \frac{1}{3}$.	$a_1 = \frac{1}{4}$.	$a_1 = \frac{1}{5}$.	$a_1 = \frac{1}{6}$.
0	0	0	0	0
30	0.644	0.609	0.587	0.572
60	1.010	0.977	0.954	0.939
90	1.000	1.000	1.000	1.000
120	0.722	0.755	0.778	0.793
150	0.356	0.391	0.413	0.428
180	0	0	0	0

To get from this a curve of piston speeds lay off *ab*, *bc*, *cd*, etc. (Fig. 130), on *AB*, each representing 30°. Erect ordinates from the table for the appropriate rod to crank ratio. From *a* lay off *aX* to represent the velocity of the crank-pin in feet per minute. Project the ordinate for 90° *dd'* across to the left, getting point *d''*. Join *d''* and *X*. Then from any other point *b'* project across and draw parallel to *Xd''* the line *b''b'''*. The distance *ab'''* will be the piston velocity at crank-angle *b* to the same scale that *ax* represents linear velocity of crank-pin.

85. Flat Valves.—Let Δ = diameter of the opening of the valve-seat, called valve diameter, in inches (neglecting the stem);

h = valve-lift, inches;

V_g = velocity of gases through the valve;

The area of the valve opening is $\frac{\pi\Delta^2}{4}$. When partially open the effective area is that of a cylindrical surface $\pi\Delta h$ square inches if the seat be flat, but less if the seat be conical.

Putting these equal and solving for h,

$$h=\frac{\Delta}{4}; \quad . \quad . \quad . \quad . \quad . \quad (120)$$

that is, the valve is full open at a lift equal to one-fourth its diameter.

If the pressure and temperature be kept constant, the velocity of the gases through the valve to the velocity of the gases entering the cylinder would be inversely as the effective areas. With flat-seated valve full open

$$\frac{S'}{V_g}=\frac{\frac{\pi\Delta^2}{4}}{\frac{\pi d^2}{4}}=\frac{\Delta^2}{d^2}. \quad . \quad . \quad . \quad . \quad (121)$$

With flat-seated valve partly open

$$\frac{S'}{V_g}=\frac{\pi\Delta h}{\frac{\pi d^2}{4}}=\frac{4\Delta h}{d^2}. \quad . \quad . \quad . \quad (122)$$

Hence

$$V_g=\frac{d^2S'}{4\Delta h}, \quad . \quad . \quad . \quad . \quad (123)$$

the maximum value of which is

$$(\text{max.})\ V_g=\frac{d^2S'}{\Delta^2}. \quad . \quad . \quad . \quad (124)$$

86. Cone-seat Valves.—The above applies to flat-faced valves. When, however, the valve is conical the gases will be discharged

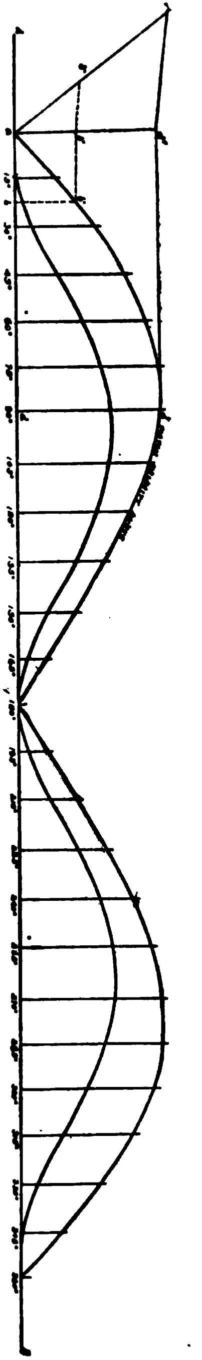


Fig. 180.

through a space between two cones and the width of opening h' will be to the lift as $\frac{1}{\sqrt{2}}$, the side to the hypothenuse of a right-angled triangle, Fig. 131. Hence

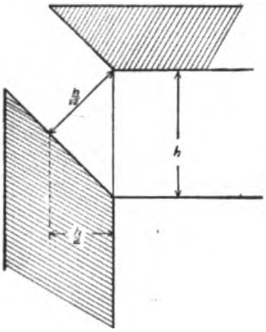


Fig. 181.

$$h' = \frac{h}{\sqrt{2}} \quad . \quad . \quad . \quad . \quad (125)$$

for a 45° angle and small lift.

Gases passing this opening will have a maximum velocity at the inner edge. This area is the surface of a truncated cone whose element has a length h' and whose small diameter is Δ and large diameter $\Delta + 2\left(\frac{h}{2}\right)$, and therefore whose mean diameter is

$$\frac{\Delta + \Delta + h}{2} = \Delta + \frac{h}{2}.$$

The area of opening is then

$$\pi\left(\Delta + \frac{h}{2}\right)h' = \pi\left(\Delta + \frac{h}{2}\right)\left(\frac{h}{\sqrt{2}}\right) = \frac{\pi \Delta h}{\sqrt{2}} + \frac{\pi h^2}{2\sqrt{2}}. \quad . \quad . \quad (126)$$

This is true until the lift is such that h' is equal to the width of the seat, after which the opening approaches that of the flat-faced valve.

The ratio of velocities will then become

$$\frac{S'}{V_g} = \frac{\pi\left(\frac{\Delta h}{\sqrt{2}} + \frac{h^2}{2\sqrt{2}}\right)}{\pi\frac{d^2}{4}} = \frac{4}{\sqrt{2}}\frac{\Delta h}{d^2} + \frac{\sqrt{2}h^2}{d^2};$$

hence

$$\frac{S'}{V_g} = 2.84\frac{\Delta h}{d^2} + 1.4\frac{h^2}{d^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (127)$$

In an engine the only quantities that vary throughout the stroke are S' and h; hence to keep V_g constant h should be proportional to S' for flat-seated valves and is given by

$$h=\frac{d^2}{4\Delta}\frac{S'}{V_g}, \quad \ldots\ldots\ldots \quad (128)$$

and the maximum useful h is given by equation (120) and S' is given by the curve of Fig. 128. From this curve the contour of the cam to keep the gas velocity constant for all points of the stroke can be obtained if used with the known area for each valve-lift.

87. Cam-curve for Constant Gas Velocity.—Cam-shafts rotate at half the speed of the crank-shaft; therefore a valve that is to be open for one stroke should have a contour changing from zero to zero

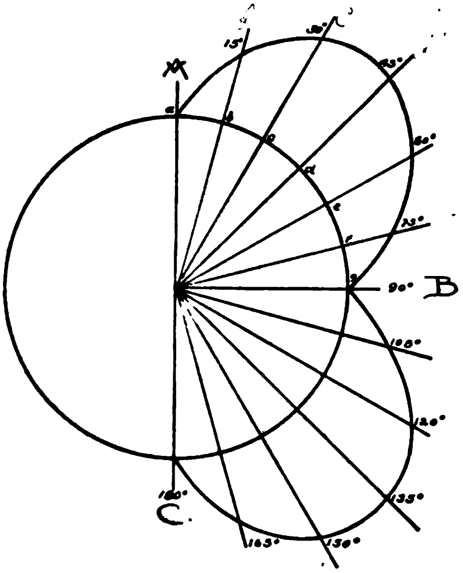


Fig. 132.

in 90° of rotation of the cam-shaft. To draw the cam contour for a constant gas velocity, lay off a circle of any radius with center O, Fig. 132. Between two diameters at right angles, OA and OB, lay off the arcs ab, bc, etc., by dividing 90° into any number of equal parts, say every 15°. Lay out bb', cc' each equal to an ordinate of the piston-speed curve for twice the angle. This curve is calculated just as was the piston-speed curve by the use of similar triangles, making *ax*, Fig. 130, equal to the maximum lift. It will appear that cams working on an outstroke (suction) should have a different contour, or rather the same with ordinates or radii reversed, than should a cam working on a return stroke (exhaust). Cams should be designed on this theoretic curve as a basis, but there may be certain variations desirable from considerations of practice. However, any change of contour from this means a variable gas velocity with stroke, except for the beginning of exhaust, when the terminal pressure must first fall to atmosphere.

88. Valve-lifts.—Valve-lifts are never given the full height of $\frac{1}{4}$ diameter for several reasons: pound on cam if the lift is high and quick, pound on the valve-seat and difficulty of closure by springs at high speeds against the inertia of the valve, coupled with the possibility of securing low velocities by larger diameter and small lifts. High-speed automobile engines use valve-lifts.

$$h_1 = 0.05\Delta \qquad (129)$$

to $$h_2 = 0.10\Delta \text{ for flat valves,} \qquad (130)$$

and $$h_3 = 0.07\Delta \qquad (131)$$

to $$h_4 = 0.16\Delta \text{ for conical seats.} \qquad (132)$$

89. Valve Diameters and Velocity Ratios.—Their diameters vary also considerably, running from

$$\Delta_1 = 0.30d \qquad (133)$$

to $$\Delta_2 = 0.45d \text{ indiscriminately.} \qquad (134)$$

Using these values in the formula the ratio of velocities for point of maximum opening can be obtained.

For flat valves:

1. $h = 0.05\Delta = 0.015d$ when $\Delta = 0.30d$.
2. $h = 0.10\Delta = 0.030d$ " $\Delta = 0.30d$.
3. $h = 0.05\Delta = 0.0225d$ " $\Delta = 0.45d$.
4. $h = 0.10\Delta = 0.045d$ " $\Delta = 0.45d$.

This gives for the four limits

Δh.	Max. $\left(\frac{V_g}{S'}\right)$ = ratio of $\frac{\text{gas velocity}}{\text{piston velocity}}$ at maximum opening and maximum piston speed.
1. $0.0045d^2$	55.55
2. $0.0090d^2$	27.77
3. 0.010125	24.69
4. $0.02025d^2$	12.345

Applying the same treatment for comparison to conical-valve limits, for conical 45° valves

1. $h=0.07\Delta=0.021d$ when $\Delta=0.30d$.
2. $h=0.16\Delta=0.048d$ " $\Delta=0.30d$.
3. $h=0.07\Delta=0.032d$ " $\Delta=0.45d$.
4. $h=0.16\Delta=0.072d$ " $\Delta=0.45d$.

The ratio of velocities becomes, for the four limiting cases, to point where piston speed and lift are maximum:

1. $\left(\frac{S'}{V_g}\right)=.0223, \quad \left(\frac{V_g}{S'}\right)=44.84.$
2. $\left(\frac{S'}{V_g}\right)=.0432, \quad \left(\frac{V_g}{S'}\right)=23.15.$
3. $\left(\frac{S'}{V_g}\right)=.0419, \quad \left(\frac{V_g}{S'}\right)=23.86.$
4. $\left(\frac{S'}{V_g}\right)=.0972, \quad \left(\frac{V_g}{S'}\right)=10.29.$

Large engines have valve-lifts about the same ratio of the diameter as do these small valves, but with their slow speeds the lift may be greater without so great a danger of harm if the valve be plain. In large valves, which are usually water-cooled and heavy, the weight including considerable water, the inertia increases with the weight about as the square of the diameter as the inertia decreases as the square of the speed. Theoretically valves of precisely the same strength and form would increase in weight with the cube of their diameters, but the form changes with size. Diameters of valves in large engines are generally relatively smaller than in the smaller high-speed engine, since the gas inertia is negligible.

90. Mean Gas Velocity.—The common method of giving these gas velocities would be to state that the gas velocity was so many feet per minute computed on assumed constancy of velocity throughout the stroke. This is called mean gas velocity and calculated from mean piston speed and continued full open valve. Thus M. F. Gaillardet gives for engines of the small high-speed class:

$$V_g = 13200 \text{ ft. per minute} \quad . \quad . \quad . \quad . \quad . \quad (135)$$

$$\text{to} \quad V_g' = 19800 \text{ ft. per minute} \quad . \quad . \quad . \quad . \quad . \quad (136)$$

and limits the lift to $\frac{1}{8}''$. This is for suction vacuum, opened or automatic valves. When at high speeds, unless the large vacuum belonging to these velocities is secured, the valves with their high inertia will not open and close promptly. Mr. Heseldin gave some time ago for automobile engines:

Exhaust.	Automatic Inlet.	Cam Inlet.
$V_g = 4500$	3600	5400

Capt. C. C. Longridge gives V_g for inlet 4800 and exhaust 6000. These last two show a not very close agreement with the first, and the reason is that in the one case a prompt automatic valve movement is sought, while in the other the avoidance of loss and as high M.E.P. as possible with correspondingly high economy is desired. Experiments by the writer show that with mechanically operated valves the loss shown by indicator cards is about 10%, with $V_g = 4000$ for both valves, and it increases about as the square of the speed. Güldner gives $V_g = 4920$ as the maximum *mean* gas velocity allowable. Very different actual velocities may give the same means, and this is a source of danger to designers.

91. Ratio of Maximum to Mean Gas Velocity.—The maximum piston velocity is to the mean velocity as crank-pin velocity is to mean piston speed or the velocity of a point on the circumference of a circle to the velocity of a point across the circumference, which gives

$$\frac{S'}{S} = \frac{\pi d}{2d} = 1.57. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (137)$$

Thus for an automobile engine $6\times6\frac{1}{2}$ at 700 R.P.M., with valves 2″ diameter flat and $\frac{5}{16}$″ lift and rod=4 cranks, the actual maximum and mean may be determined.

Here

$$S=758=\frac{2\times6\frac{1}{2}\times700}{12},$$

$$S'=\frac{\pi\times6\frac{1}{2}\times700}{12}(\sin\theta+\tfrac{1}{8}\sin 2\theta),$$

which is a maximum at 90° giving

$$S'=\frac{\pi\times6\frac{1}{2}\times700}{12}$$
$$=1.57\,S=1192.$$

The mean gas velocity, as usually used, is given by the formula

$$V_g=S\frac{d^2\pi}{4\Delta l}=758\times7.2.$$

The maximum is

$$V'=1192\times7.2=1.57V,$$

provided this maximum occurs when valve opening and piston speed are both at their maximum.

Should the ratio of $\frac{V'}{S'}$ differ at various points of the stroke, there will be a variable resistance to the motion of the gases and possibly considerable loss for the few changes of motion per minute, which is only about 0.10 that of small engines, reducing gas velocity to 0.01, its value for the small engines. But a desire for smaller losses causes, in many cases, the larger ratios to be retained. It will be found that for these large engines

$$\Delta_3=0.25d \quad . \; . \; . \; . \; . \; . \; . \; . \quad (138)$$

$$\text{to} \quad \Delta_4=0.50d, \quad . \; . \; . \; . \; . \; . \; . \; . \quad (139)$$

the smaller values being quite practicable, but the larger values involving less loss by too high a gas velocity. The majority of cases will run about

$$\Delta_5=0.32d. \; . \; . \; . \; . \; . \; . \; . \; . \quad (140)$$

It is recommended that wherever possible Δ be made larger than this.

92. Contracted Vein at Valve Opening.—So far only the extent of the opening has been considered and the velocity obtained by dividing the quantity of gas passing by the area of the passage. It is well known, from hydraulic experience chiefly, how seriously the effective area of an opening can be lowered from the actual. Sharp-edged openings will discharge only a fraction of what may pass round edge openings of equal areas with nicely curved approach and discharge surfaces. Care must be taken, then, to make approaches rounded, and all sharp edges should be eliminated, else the actual gas velocity will be higher than above calculated. Valve ports and passages should be so large that the gas velocity shall not exceed 3000 feet per minute, especially if frequent bends and sharp corners are necessary. Especially important is it for exhaust passages near the engine to have ample area and proper form, for immediately on lifting the valve a rapid rush of gas takes place, while the pressure drops to atmosphere, and if this drop be not complete, a back-pressure line will show on the indicator card, calling for a more advanced opening. Both these last two points have not always received from builders the consideration they merit, but when high economy and high mean effective pressure are desired they assume great importance.

93. Valve Thickness.—When small, and even too when large and on the suction side, poppet-valves are flat discs with a stem attached. Economy of metal and elegance of design bring about an arching of the valve which, when the valve is large and hollow for water-cooling, becomes very marked, approaching a hemisphere. On the side where the stem is attached there is usually a small rounded fillet for strength, but in some cases of good design this fillet is large, extending by an arc of circle tangent to stem and valve face right to the edge of the valve. This furnishes an excellent guide for the gas flow, and in connection with similar reversed roundings of the walls makes the actual opening more effective by reducing the percentage of contraction in the contracted vein. Large valves are sometimes balanced, but this balancing serves only to reduce the stress on the stem of an exhaust-valve on the initial opening and affects not at all the load on the crown of the valve.

The proper thickness of a valve disc can be judged most easily by considering the valve a uniform flat plate supported, not fixed, at the edges. When rounded less metal is necessary and least when hemispherical. Such a flat plate is a kind of beam subject to flexure and in these cases uniformly loaded. The tension of the closing spring increases the load, but by so small an amount compared with the maximum gas pressure that it may be neglected. For such flat plates Grashof gives

$T=\sqrt{\frac{5}{6}\left(\frac{\Delta}{2}\right)^2\frac{P_m}{f}}$, formula (93) applied to a valve of diameter Δ;

where T = the thickness of plate in inches;
t_v = the thickness of valve disc;
Δ = diameter of valve receiving unbalanced pressure;
f = working stress in the metal;
P_m = maximum uniform pressure pounds per square inch.

From this

$$t_v=0.45\Delta\sqrt{\frac{P_m}{f}}. \qquad \text{(141)}$$

Small engines have thickness

$$t_v'=0.09\Delta \qquad \text{(142)}$$

$$\text{to} \quad t_v''=0.23\Delta. \qquad \text{(143)}$$

Hence for these cases we have

$$f'=25P_m \qquad \text{(144)}$$

$$\text{to} \quad f''=4P_m, \qquad \text{(145)}$$

showing a very wide range, the lower values being very much too low for strength alone, but giving a very stiff disc. In fact a valve that springs appreciably will not only be impossible to keep tight but will in time surely break.

Taking P_m at 300 and 450 lbs. the stresses become for the four cases

$$\begin{aligned} f' &= 0.25P_m = 7500, && \text{when } P_m=300,\\ &= 11250, && \text{``}\ P_m=450;\\ f'' &= 4P_m = 1200, && \text{``}\ P_m=300,\\ &= 1800, && \text{``}\ P_m=450. \end{aligned}$$

Güldner recommends for cam-moved valves

$$f''' = 5700, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (146)$$

but with automatically opened suction-valves, which must be extremely light,

$$f^{\mathrm{iv}} = 11000. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (147)$$

Hollow valves frequently have crowns $0.40d$ above the base with a thickness of metal about $0.07d$ when properly arched.

Solid valves may be arched by tapering off edges to 0.6 the center thickness and frequently the edge is beveled or rounded.

Just at the opening an exhaust-valve is subject to different stress than before. At this time the gas pressure continues to act as before, while the support has changed from an edge support at seat to a central one by the stem; but a valve strong enough for the former stress will be more than strong enough for the latter.

94. Valve-faces and Valve-seats.—Valve-faces depend for their width on considerations of tightness and strength, for sufficient pressure must be exerted to keep the joint tight and yet not crush or otherwise injure the metal of the seat.

The face of a flat valve is an annulus, while the active pressure-resisting surface is the area of face of the seat. The seat is raised from surrounding metal and the valve is large enough to project beyond.

Conical valves have faces wider than the seat, the projection being upward to permit of regrinding. In this case also the effective supporting face is that of the seat.

$$\text{Width conical valve-face} = \left\{ \begin{matrix} 1.1, \text{ width of seat, to} \\ 1.5 \quad \text{"} \quad \text{"} \quad \text{"} \end{matrix} \right\} . \, . \quad (148)$$

95. Seats for Flat Valves.—Let f_v = width of seat in inches, P_v = the desired or allowable compression on the joint in pounds per square inch.

In calculating the total load on the valve the outer circle of the seat should be taken, as the inner circle will probably wear most and leave the whole seat active in receiving stress from above.

This makes the load

$$P_m \times \frac{\pi(\Delta+2f_v)^2}{4} \text{ pounds.}$$

This is resisted by the compression on the ring of

$$P_v \times \left[\frac{\pi(\Delta+2f_v)^2}{4} - \frac{\pi\Delta^2}{4}\right] \text{ pounds.}$$

Hence

$$(\Delta+2f_v)^2[P_v - P_m] = \Delta^2 P_v.$$

$$\therefore\ \Delta+2f_v = \Delta\sqrt{\frac{P_v}{P_v-P_m}},$$

or

$$f_v = \frac{\Delta}{2}\left(\sqrt{\frac{P_v}{P_v-P_m}}-1\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad (149)$$

Flat valves have faces varying from

$$f_v' = 0.04\Delta, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (150)$$

giving $P_v = 7.25P_m$,

$$\text{to } f_v'' = 0.09\Delta, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (151)$$

giving $P_v = 3.56\ P_m$.

With $P_m = 300$, $f_v = 0.04\Delta$ gives $P_v = 2175$ lbs. sq. in.
$P_m = 300$, $f_v = 0.09\Delta$ " $P_v = 1068$ " " "
$P_m = 450$, $f_v = 0.04\Delta$ " $P_v = 3263$ " " "
$P_m = 450$, $f_v = 0.09\Delta$ " $P_v = 1602$ " " "

This maximum pressure, it must be remembered, lasts only a short time and on a poor seat may not be equally distributed, as here assumed. Valve-seats must be strong enough to carry this rim concentrated load without springing, or more particularly, without *unequally* springing.

96. Cone-valve Seats. — The normal pressure on cone-seated valve-faces is greater for equal diameters and width of seat than for flat-seated valves, and while not strictly true, it is near enough for this purpose to take the normal pressure for cone-seated valves P_v' as

$$P_v' = \sqrt{2}\, P_v = 1.41\, P_v \text{ (for } 45^\circ).$$

Practice has sanctioned quite some difference in the width of a valve-seat for cone-valves also. Small engines have cone-seats of widths from

$$f_v''' = 0.09d, \quad \ldots \ldots \quad (151)$$

giving $P_v = 5.02\, P_m$, approximately,

$$\text{to} \quad f_v^{\text{IV}} = 0.15d, \quad \ldots \ldots \quad (152)$$

giving $P_v = 3.47\, P_m$, approximately.

Large engines usually have smaller seats proportionally, though not always. For these the width of seat will vary from

$$f_v^{\text{V}} = 0.06d \quad \ldots \ldots \quad (153)$$

$$\text{to} \quad f_v^{\text{VI}} = 0.10d. \quad \ldots \ldots \quad (154)$$

The outside diameter of the valve is a matter of no particular consequence, but it should be large enough to permit of the face overlapping the seat.

97. **Valve-stems.**—Exhaust-valve stems must be strong enough to lift the valve with the full gas pressure at release. This terminal pressure can be computed from the pressure ratio for the fuel applied to atmospheric pressure for full expansions. When, however, ignition is late the terminal pressure may be 1.5 times as high. It will suffice if stems are amply strong to resist this latter, which will range from

P_t = terminal cylinder pressure = 2.2 atmospheres for weak fuel, high compression, and early ignition

to P_t = terminal cylinder pressure = 6.5 atmospheres for rich fuel, low compression, late ignition, and full charges.

The latter high valves are met only in spark control gasolene automobile engines. For common stationary work on proper ignitions, an easy figure to keep in mind is 50 lbs. per square inch; this of course is very *indefinite.*

If the stem were free and of a length exceeding 15 diameters it would be treated as a column and so subject to flexure before crushing. This would give heavy stems. It is usually possible and desirable to break the stem and introduce other elements of the valve-gear to transfer the cam motion to the valve stem so that it shall not act as a column. The stem is carried in a stem-bearing varying in length from

$$\text{Stem-bearing length} = 1.4\Delta \quad . \quad . \quad . \quad . \quad . \quad . \quad (155)$$

$$\text{to} \quad \text{''} \quad \text{''} = 3.5\Delta. \quad . \quad . \quad . \quad . \quad (156)$$

This bearing guides the stem over more than half of its length and prevents flexure. These considerations enable stems to be calculated on a basis of compression chiefly. It is, moreover, desirable, because of difficulty of lubrication and hence wear, to provide excess diameter of stem in engines, where weight is no consideration, to permit of re-turning and insertion of a bushing.

The total load on the stem is, at the moment of opening the exhaust-valve,

$$P_t \times \pi \frac{(\Delta + 2f_v)^2}{4}.$$

Calling C the compression stress in the metal and m the stem diameter,

$$C \times \frac{\pi m^2}{4}$$

is the resistance. Hence

$$C = P_t\left(\frac{\Delta + 2f_v}{m}\right)^2,$$

$$\text{or} \quad m = (\Delta + 2f_v\sqrt{\frac{P_t}{C}},$$

and $2f_v$ varies from 0.08Δ to 0.18Δ for flat valves. Hence

$$m = 1.08\Delta\sqrt{\frac{P_t}{C}} \quad \text{to} \quad m = 1.18\Delta\sqrt{\frac{P_t}{C}}.$$

In practice, valve-stems have diameters in small engines with no intended reboring of

$$m_1=0.22\Delta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (157)$$

$$\text{to} \quad m_2=0.28\Delta. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (158)$$

This gives for the four limiting cases:

$$\left.\begin{array}{lllll} 1.\ \text{When} & f=0.08\Delta & \text{and} & m=0.22\Delta & C=24P_t \\ 2.\quad `` & f=0.18\Delta & `` & m=0.22\Delta & C=29P_t \\ 3.\quad `` & f=0.08\Delta & `` & m=0.28\Delta & C=15P_t \\ 4.\quad `` & f=0.18\Delta & `` & m=0.28\Delta & C=18P_t \end{array}\right\}\text{Approximately.}$$

And when $P_t=50$ lbs. sq. inch the four cases give stress:

1. $C=1200$ lbs. per sq. in.
2. $C=1450$ " " " "
3. $C=\ \ 750$ " " " "
4. $C=1900$ " " " "

These stresses are properly low, for exhaust-valve stems work very hot; large diameters also help to conduct away heat better.

Güldner gives for stem diameter $m=0.12\Delta+\frac{3}{16}''$ to allow for reboring. Practice in large stem diameters will lie between the limits of

$$m_3=0.22\Delta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (159)$$

$$\text{and} \quad m_4=0.35\Delta, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (160)$$

the smaller value being amply strong, but the larger one admitting of inside boring for water-cooling and the removal thus of half a diameter and leaving for the thickness of the walls of the stem 0.09Δ, approximately; in fact a stem 0.24Δ will permit of similar interior boring and still be strong enough. If half diameter be bored the metal removed has an area $\dfrac{\pi\times(0.12)^2}{4}\Delta=1.131\Delta$ sq. inches. Originally the area was $\dfrac{\pi\times(0.24)^2}{4}\Delta=4.5239\Delta$ sq. inches, leaving 3.3929Δ sq. inches, which corresponds to

$$m'=0.21\Delta.$$

It is well to check a stem by the column formula when the bearing is not sufficient to restrain bending or the stem unusually long.

Stems on inlet valves may be of practically any thickness consistent with overcoming (*a*) the inertia of the valve in opening, during which time the stem is in compression; (*b*) the necessary tension of the spring. The stem must also guide the valve properly to the seat. None of these stresses is large. If the diameter be made half that for exhaust stems ample strength will be secured.

98. Valve-closure Springs.—Practically no mechanically closed poppet-valves are used, spring closure being the rule. The tension of the spring must be such as will close the valve against (*a*) its inertia, and possibly also some linkage inertia, (*b*) stem friction, and possibly some link friction, and (*c*) gas friction, sometimes called windage or fluid friction.

Helical coil springs are almost universally used because of their adaptability to the stem directly, and these are generally cylindrical helices, though a cone base is sometimes used. In these coil springs the stress is purely torsional.

The first solution in a valve-spring problem determines the force necessary to close the valve in a given time, or the necessary permanent load W_s, which will not change much for the lifts used, which are small compared to the initial deflection necessary to produce the axial load of W lbs. To keep spring tension from varying much with deflection, many turns are used, about 8 to 15 for cam-moved valves, against 5 or 6 for automatic valves. Similarly the valve gravitational weight is neglected as assistance or resistance in ~~its~~ closure. A valve weighing z pounds must move through $\frac{h}{12}$ feet in T seconds.

The acceleration is $\frac{2h}{12\times T^2}$ ~~inch~~ ft second-units.

$$\therefore\ W_s=\frac{zh}{32.2\times 6\times T^2}=\frac{zh}{193.2\times T^2} \quad . \quad . \quad . \quad (159)$$

=pounds to close the valve in T seconds.

Or,
$$T=0.072\sqrt{h\frac{z}{W_s}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (160)$$

seconds will be necessary for the closure if friction = 0.

99. Period of Closure.—A spring should be capable of closing a valve in ¼ of a stroke or ⅛ of a revolution at most, as this is a little less than the time from the crest of the piston-velocity curve to the end of the stroke. As one revolution takes $\frac{1}{N}$ minutes or $\frac{60}{N}$ seconds, this time becomes

$$T=\frac{60}{8N}. \quad \therefore T^2=\frac{56.25}{N^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (161)$$

Substituting above the necessary tension for ¼ stroke closure we have

$$W_s=\frac{zhN^2}{10867.5} \text{ pounds.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (162)$$

This subject of the part of stroke to which to limit valve-closure has been investigated for automatic valves on automobile engines by Mr. H. L. Towle. He finds for a number of cases that this quantity, whose maximum value has been set at

$$\frac{\text{Time to close valve}}{\text{Time per stroke}}=0.25,$$

varies from

$$\text{Minimum } \frac{\text{Time to close valve}}{\text{Time per stroke}}=0.043 \quad . \quad . \quad . \quad . \quad (163)$$

$$\text{to Maximum } \frac{\text{Time to close valve}}{\text{Time per stroke}}=0.234, \text{ with} \quad . \quad . \quad . \quad (164)$$

$$\text{Average } \frac{\text{Time to close valve}}{\text{Time per stroke}}=0.134. \quad . \quad . \quad . \quad . \quad (165)$$

100. Spring Tension.—The spring tension per square inch of valve will vary from 6 to 30 oz., the mean being about 12 oz., or 0.75 lb., for automatic- or vacuum-opened valves. Güldner recommends tensions from 0.70 to 1 lb. per sq. inch of valve, depending on the speed and valve weight. For mechanically operated valves the tension increases and ordinarily lies between the limits of

$$W_s' = 5.5\frac{\pi \Delta^2}{4} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (166)$$

and

$$W_s'' = 9.0\frac{\pi \Delta^2}{4}. \quad . \; . \; . \; . \; . \; . \; . \; . \quad (167)$$

These values for the force or spring tension to close the valve in some predetermined fraction of the stroke should always be calculated from the formula given and to the result a reasonable amount added for stem friction and windage. The result should lie within the empirical values given above.

101. Spring Dimensions.—The second part of the spring-closure problem determines the dimensions of the spring on the data of tension needed to bring about appropriate closure. For this purpose two formulas are needed. The first gives the relation between compression of the coil in inches and the load to produce it, while the second gives the maximum safe working load, both in terms of the dimensions of the coil.

Let d_w = diameter of wire in inches;
d_c = diameter of coil in inches;
G = modulus = 12,000,000 for steel;
i = compression of coil in inches;
n_c = number of turns in coil;

then

$$W^s = \frac{G d_w^4 i}{8 d_c^3 n_c} = \frac{1{,}500{,}000 d_w^4 i}{d_c^3 n_c}, \quad . \; . \; . \; . \; . \; . \quad (168)$$

and

$$\text{Safe } W_s' = \frac{27500 d_w^3}{d_c} \text{ for wire } \tfrac{1}{4} \text{ inch diameter} \quad . \; . \; . \; . \quad (169)$$

$$= \frac{23500 d_w^3}{d_c} \text{ `` `` } \tfrac{3}{8} \text{ `` `` } \quad . \; . \; . \; . \quad (170)$$

$$= \frac{19600 d_w^3}{d_c} \text{ `` `` } \tfrac{1}{2} \text{ `` `` } \quad . \; . \; . \; . \quad (171)$$

$$= \frac{40000 d_w^3}{d_c} \text{ `` `` smaller than } \tfrac{1}{4} \text{ inch diameter.} \quad (172)$$

The tension desired should be substituted in the formulas (169) to (172) to first determine the safe size of wire, but this requires a knowledge of the coil diameter.

Small engines use coils of quite large diameter compared with the valve disc, varying from

$$d_c=0.7\Delta \quad \text{. . . . .} \quad (173)$$

$$\text{to} \quad d_c'\cancel{D}=1.1\Delta. \quad \text{. . . . .} \quad (174)$$

As the size of spring wire does not differ so much with the size of engine, large engines have coils whose diameters range from

$$d_c''=0.3\Delta \quad \text{. . . . .} \quad (175)$$

$$\text{to} \quad d_c'''=0.8\Delta. \quad \text{. . . . .} \quad (176)$$

The diameter of wire thus determined as safe may be used or a larger diameter, or, better still, that next larger diameter that corresponds to some number on a standard wire-gauge scale. Having the size of wire and diameter of coils with the load required, the number of inches initial compression on the spring to give this value can be found by substituting in formula (168) if the number of turns needed be known.

For automatic vacuum-opened valves

$$n_c'=5 \quad \text{. . . . .} \quad (177)$$

$$\text{to} \quad n_c''=8 \quad \text{. . . . .} \quad (178)$$

will be within good practice. Cam-opened valves have more turns, the number lying between

$$n_c'''=6 \quad \text{. . . . .} \quad (179)$$

$$\text{and} \quad n_c^{\text{IV}}=15. \quad \text{. . . . .} \quad (180)$$

For convenience of computation a wire-gauge scale is given at the end of the book.

There is in this spring practice a possible and legitimately wide range of dimension. The only real limitations based on the needs of the valve are

1°. A certain permanent minimum load.

2°. No great change in the load for the valve-lift.

The first results from using metal sufficiently thick in a coil of a certain diameter and giving the coil an initial compression; the second from using a sufficient number of turns to the coil, which will vary, with the judgment of the designer, from 15 to 8 in large engines. Where the conditions permit, a long coil is best giving more uniform load, least permanent set, and should have a good diameter.

102. Actual Cam-curves for Valve Opening and Theoretic cam-curve.—Valves are opened by (*a*) the difference in pressure in the cylinder from atmosphere acting against the inertia and small spring tension of the valve, or (*b*) by cams and linkage systems. When cams are used their form is a matter of considerable importance. The manner in which the valve should be moved to secure a constancy of velocity of gases through the opening from the piston-speed curve has been explained, but the actual cam curve is not the same as the actual piston-velocity curve.

1°. A roller contact with the cam face must produce the motion of the stem desired and the cam face must be tangent to the roller circles when the roller centers describe the desired movements.

2°. The cam and roller do not always actuate the valve-stem directly; frequently cranks, levers, tension-rods, struts, and pins constitute a linkage system to transfer the motion from the place where it is convenient to put the cam to the place where it is desirable that the valve-stem should be. There is also lost motion or intentional clearance always present between cam and roller, else the valve might not close.

3°. The actual valve settings are never the same as are theoretically desirable. Valves do not open or close exactly at the end of the stroke, be they inlet or exhaust.

4°. Different considerations apply to the exhaust-valve than to the suction-valve because the functions of the two are different. On release, there must be allowed a sufficiently large opening and free passage beyond to permit enough gas to pass to allow the terminal pressure to drop to atmosphere before the exhaust-stroke proper begins. If temperature and pressure did not change and terminal pressure were three atmospheres, practically twice as much gas would exhaust during that first release period at the end of the expansion-stroke as during the whole following exhaust-stroke proper. These two exhaust periods will be called the first-period exhaust, which is largely a part of the expansion-stroke, and second-period exhaust, which is the exhaust-stroke proper.

These four reasons why the actual cam-curve differs from the piston-velocity curve affect the cam-curve in the following ways:

The first cause affects the curve form by an amount depending on the diameter of the roller, a zero-roller diameter having no effect and a large-diameter roller greatest effect.

The second cause of difference affects merely the amplitude of the cam-crest above zero-circle compared to the curve for the valve-stem. If there were no linkage these would be equal, and they may even be equal with linkage; but where linkage causes the path of the roller center and valve to be unequal, then will cam-crest be of different height above the zero-circle than is the crest of the valve-lift curve above its closed position, zero-line.

The third cause affects the form and height of the cam-curve little or not at all, but calls for differences in the points of beginning and ending of the curve. It determines, in short, the lag and lead of the valve.

The fourth cause of variation results in a large lead for the exhaust-valve, with a possibly quicker rise in the curve for exhaust-cams if their inertia will permit.

103. Cam-roller Effect.—Lay off the piston-speed curve for 180° on a base (Fig. 133) similar to Fig. 130, to such a scale as will give the maxi-

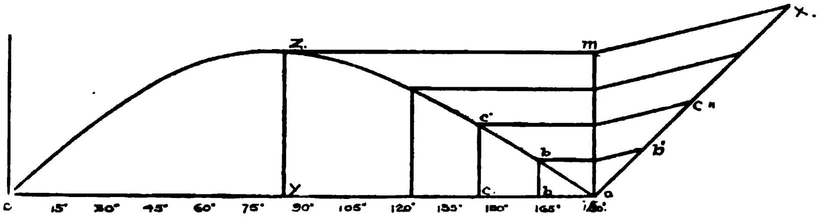


Fig. 133.

mum ordinate *yz* equal to the valve-lift. Transfer this to a circular base of double angles, getting thus the whole curve in 90° (Fig. 134), on which excess radii are desired cam eccentricities. The ordinates giving the cam eccentricities are to be obtained from the valve-lift piston-velocity curve by laying off aX to represent the maximum cam-roller center movement desired by reason of the valve motion and linkage system. Draw parallels from b' and c' to the right, joining the maximum point m with X and by drawing parallels get ordinates to proper scale, which are to be laid off on Fig. 134. The radius O_a should be the sum of OA = radius of zero-circle or hub of cam, and aA = radius of roller. With aA as radius and points b''', c''', etc., as centers draw circles, then the cam-curve drawn tangent to these circles will be that accounting for the roller influence, and by properly taking the scales for aX, will also account for motion difference in the ends of the linkage. Should clearance be desired, the radius OA should be taken as much larger than the cam-hub as the clearance desired. It should be noted that at the ends of the curve the roller coincides with the cam-curve and here will not run smooth and hence requires a flattening.

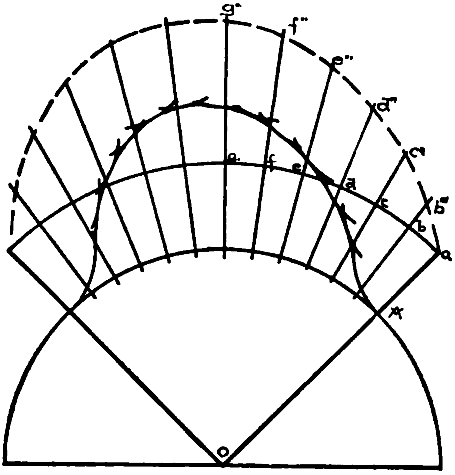


Fig. 134.

104. Ratio of Cam-lift to Valve-lift.—The ratio of valve movement to cam movement in practice will depend on the speed at which it is desired to operate on the radius of the zero-circle and on the lift. Call the cam movement h_c and valve-lift h, radius of zero-circle of cam r_c, and radius of roller r_r.

In small high-speed engines the motion is usually direct and

$$h_c = h. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (177)$$

In cases like this radius of zero cam-circle is frequently the same as that of the cam-shaft. The relation between h_c and r_c is usually about

$$r_c = 1.5h_c, \quad \ldots \ldots \quad (178)$$

though it may go as high as

$$r_c' = 2.8h_c \quad \ldots \ldots \quad (179)$$

and as low as

$$r_c'' = 0.8h_c. \quad \ldots \ldots \quad (180)$$

In these cases the rollers are usually of radius

$$r_r = 0.8r_c, \quad \ldots \ldots \quad (181)$$

the limits being

$$r_r' = 1.2r_c \quad \ldots \ldots \quad (182)$$

and

$$r_r'' = 0.6r_c. \quad \ldots \ldots \quad (183)$$

Medium-speed stationary engines have cam-lifts varying from

$$h_c' = 0.5h \quad \ldots \ldots \quad (184)$$

$$\text{to} \quad h_c'' = 1.0h. \quad \ldots \ldots \quad (185)$$

105. Zero Radius of Cams.—The cams for these stationary engines are of great variety so far as size goes, but with a zero radius of

$$r_c''' = 3.0h_c \quad \ldots \ldots \quad (186)$$

easy running will be assured, especially if the cam-roller have a corresponding radius of

$$r_r = 0.6r_c.$$

Cams are occasionally used as large as

$$r_c^{\mathrm{IV}} = 6.0h_c. \quad \ldots \ldots \quad (187)$$

These work exceedingly smooth, but as they increase the lever-arm, by which the cam-shaft lifts the valve by means of the cam, correspondingly greater stresses result. Keeping the lever-arm as small as practicable reduces the cam zero-circle to a minimum radius of

$$r_c = 1.5h_c.$$

The largest practicable roller will seldom exceed

$$r_r = 0.8r_c,$$

while the sharpest allowable contact is secured by a roller of radius

$$r_r = 0.3r_c.$$

106. Cam-curves for Proper Valve-setting.—By leaving play and adjustments in the linkage system between the cam and valve-stem, a certain kind of adjustment of period is possible. By increasing free play, a valve will open later than before and close sooner, and vice versa, but the opening period cannot be adjusted independently of the closing period. This must be done by the cam form. The above adjustment makes one phase later and the other earlier. Both may be made earlier by advancing the cam with respect to the driving-shaft. All these adjustments are available to the erector, but the designer should work as closely as his information will permit without dependence on such means when forming the cam face curves.

107. Valve-setting and Inlet-cams.—Inlet-valves should not open till exhaust-valves have closed or a back-fire may result by opening of the connection between exhaust-pipe and mixing chamber, especially with several cylinders on the same exhaust. Exhausts close on center or slightly after center, hence in the latter case inlets open still later.

Inlet Opens.	Engine.
About 5° after centre,	large slow-speed engines, 100–250 R.P.M.
" 3° " "	medium- or slow-speed small engines.
" 6° " "	small high-speed engines, 500–800 R.P.M.

Suction should continue till after center has been passed, especially in high-speed engines where the gas has considerable inertia.

Inlet Closes.	Engine.
About 10°	for large slow-speed engines.
" 4°	for medium or slow-speed small engines.
" 15°	for small high-speed engines.

To secure a retardation equal at both ends and still retain the form of the roller center-line curve the same in form as the piston velocity curve for constant admission velocity, it might seem at first glance feasible to retard the whole cam, as in Fig. 135, the full line showing the original position and the dotted line the retarded position. A glance will show that the valve-lifts will not now be proportional to the piston speed, but on the rise opening will be smaller, while on the fall larger, than before retardation at the same crank-angles.

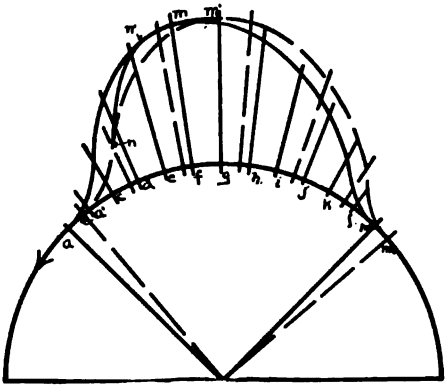


Fig. 185.

It is more important that the end of the curve give full opening with a decrease in gas velocity, perhaps, than that the rise of the curve be exact. For this reason the result of the process, while not securing the result desired, is not exactly detrimental. By making, however, a circular crest *mm'* and using the old rising curve in part and crossing over to the new one at the beginning, as shown by the dotted line *nn'*, there will result a curve with the following characteristics:

(*a*) The proper retardation of opening *aa'*.

(*b*) Constant velocity of admission *n'* to *m* for a valve with area proportional to lift.

(*c*) A higher velocity of admission *n* to *n'*.

(*d*) A lower than (*b*) velocity of admission *m'* to *m'''*.

(*e*) The proper lag of closure *m''m'''*.

To secure unequal retardation of opening and closure proceed to draw the falling line *m'm'''* by transferring the part of the curve from crest to closure. Draw a circular arc *mm'* connecting this crest with the original crest, and ease off somewhere at *n'* to the point of opening wanted. If the speed be very high at the cam-face, or the valve-spring too weak for the valve inertia, the roller may jump clear on the falling side. In such cases this crest may be eased off and still secure gas velocities lower than during the rise, or the spring tension may be increased.

To secure the lowest possible velocity of the gases the maximum lift or height of crest above the zero-circle should be $\frac{\Delta}{4}$, or one quarter of a valve diameter, neglecting the stem. Draw then Fig. 136, a curve with these data (Fig. 136), and cut off the crest by a circular arc, whose radius is the zero-circle plus the actual allowable valve-lift *dd''*. In this case from *m* to *n* gas enters with the increasing velocity and, neglecting gas inertia, from *n* to *n'* the velocity decreases, until at *n'* the velocity is the lowest possible for the valve diameter and piston

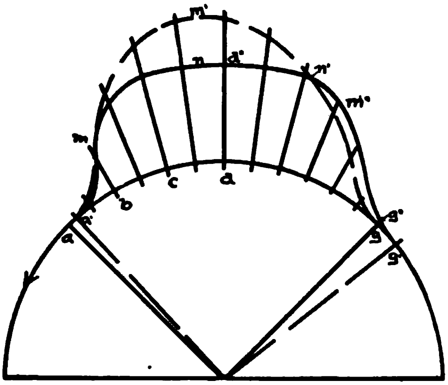


Fig. 136.

speed. The piston speed becomes zero at g'', the valve continuing open to permit the inertia of the gases to more or less raise the pressure in the cylinder to atmosphere. It will be found advisable to eliminate sharp turns at m'', as they add nothing to the suction, while it is very detrimental to the roller.

108. Example of Inlet-cam Design.—Most valves have conical seats, and all linkages have clearance, while the cam-curve is more often made for easy running than for constant velocity. The treatment of such a case with a very common form of cam consisting of two flat faces and one circular arc is best made clear by an example. One of these ordinary inlet-valves was set with an opening lag of 10° and closure on center and with a clearance of $\frac{3}{16}$ inch. It has two flat faces *ab* and *cd*, with a circular connecting arc *bc*, Fig. 137. This cam-curve is intersected by radial lines at each $7\frac{1}{2}°$ of cam-shaft rotation, beginning at the point of

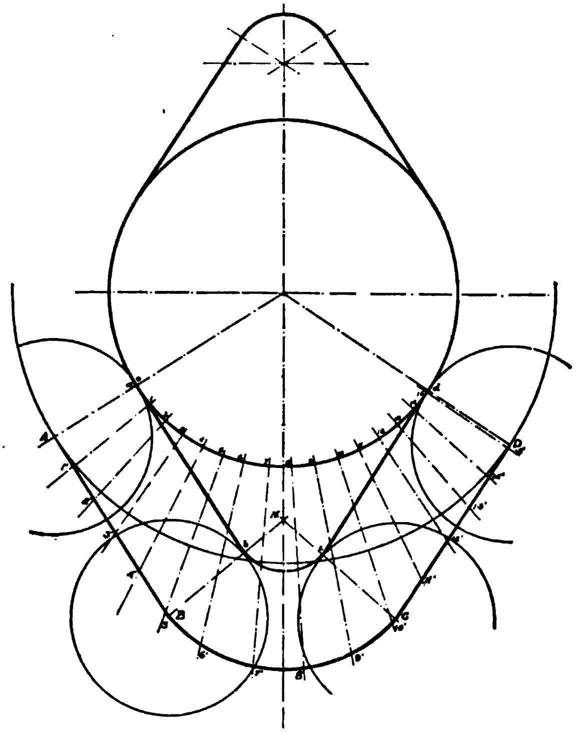


Fig. 137.

tangency of the cam-curve and cam zero-circle, and each corresponding to 15° on the crank-circle. These points are numbered. The cam-face rises above the zero-circle for fifteen full parts and approximately one fifth of one part, which is 114° (approximately), corresponding to 228° on the crank-circle. On Fig. 138 equal spaces each 1 inch long and each representing 15° of the crank rotation. The valve opening 10° after center places the zero-point on *O*, Fig. 138, and at each 15° space the ordinate represents the valve-stem lift. To get this valve-stem lift for each crank-angle lay off *a–A*, Fig. 137, to represent the cam-roller radius. *ABCD* will then be the path traveled by the cam-roller center, and hence the valve-stem for continual contact of roller with cam. The lift of the valve is given by the distance between this and the zero-circle *AMD*. This gives curve Fig. 138, which shows the valve closing at *D* 238° on the crank-pin circle. A clearance of $\frac{3}{16}$ inch between the roller and cam is indicated by moving the axis, Fig. 138, up this distance, as indicated. giving a new scale of crank-angles on which the valve when opened 10° after center closes about 180° or on the out-center.

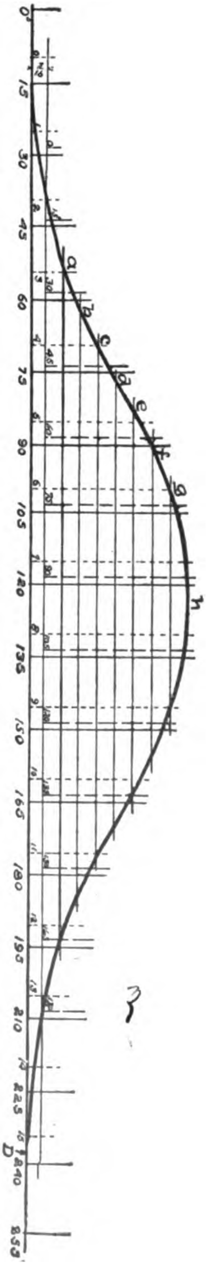


Fig. 138.

The angles at which the valve will have successively $\frac{1}{4}$ inch, $\frac{1}{2}$ inch, etc., of lift are found by laying off these distances above the clearance-line and finding the intersection with the cam-curve, Fig. 138, at the points *a*, *b*, *c*, *d*, etc.

The piston-speed curve is plotted beside the cam-curve from the following data of actual piston speeds for each crank-angle:

Crank-angle.	True Piston Speed. Feet per Minute.	Crank-angle.	True Piston Speed. Feet per Minute.
0°	.000	120°	927
30°	682	135°	726
45°	940	150°	495
60°	1113	180°	000
90°	1178		

By laying off quarter inch of valve-stem lift above the clearance-line, Fig. 138, the crank-angles for each quarter-inch lift and corresponding piston speed can be read off by reference to the scale. For this case the following numerical values were found:

Crank-angle.	Valve-lift. Inches.	Piston Speed per Minute. Feet.	Crank-angle.	Valve-lift. Inches.	Piston Speed per Minute. Feet.
25°	¼	584	114°	2	1008
34°	½	744	122°	1¾	908
41°	¾	872	130°	1½	816
48°	1	980	135°	1¼	728
55°	1¼	1048	142°	1	636
60°	1½	1112	148°	¾	528
68°	1¾	1152	156°	½	412
76°	2	1176	165°	¼	256
95°	2¼	1168			

For these piston speeds and valve-lifts the valve-opening must be found. As the valve is conical, the lift will not give the altitude of the surface of opening, nor will the mean diameter be that of the seat. The opening will be the surface of a truncated cone whose side is the least distance between valve-seat and valve-face and

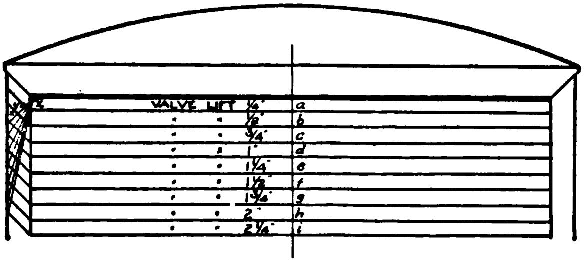


Fig. 139.

whose mean diameter is twice the distance from the center of the side of cone to the center line of the valve. These distances are most clearly found by laying down as in Fig. 139, and are given in table following. By applying the piston area the gas velocity is at once found, for the gas velocity = piston velocity × ratio of piston area to valve-opening area.

Crank-angles.	Valve-lift. Inches.	Side of Cone.	Mean Diameter	Area Open.	Gas Velocity *v*.
25°	¼	0.17	8.56	4.57	75,900
34°	½	0.34	8.66	9.25	47,600
41°	¾	0.52	8.82	14.40	35,900
48°	1	0.71	8.87	19.78	29,400
55°	1¼	0.92	8.87	25.63	24,300
60°	1½	1.14	8.87	31.76	20,800
68°	1¾	1.38	8.87	38.44	17,800
76°	2	1.62	8.87	45.13	15,500
95°	2¼	1.87	8.87	52.10	13,300
114°	2	1.62	8.87	45.13	13,200
122°	1¾	1.38	8.87	38.44	14,000
130°	1½	1.14	8.87	31.76	15,300
135°	1¼	0.92	8.87	25.63	16,900
142°	1	0.71	8.87	19.78	19,100
148°	¾	0.52	8.82	14.40	21,800
156°	½	0.34	8.66	9.25	26,400
165°	¼	0.17	8.56	4.57	33,300

The maximum possible opening is the area of the seat less that of the stem, which is 56.74 − 3.14 = 53.6 sq. ins., and this is very nearly reached with the 2¼-inch lift.

If the valve-area is always proportional to the piston speed, the gas velocity with this valve need never exceed 13,100 feet per minute if the cam-curve were parallel to piston-speed curve, except perhaps near the dead-centers, where lag of opening and closing allows piston some speed with zero valve-opening.

109. Exhaust-cams.—To permit early drop to atmosphere exhaust must open early, and the advance depends on the diameter of the valve, on freedom of passages beyond the valve, and on the speed. The following leads apply only to large full valve-diameters and should be greater for small valves.

To dispel as completely as possible the gases; if the speed be high it may hold open after center, or close on center for low speeds.

Exhaust Opens.	Engine.
25° before center.......	Slow-speed engines, 100 to 250 R.P.M.
35° before center.......	Small high-speed engines, 500 to 800 R.P.M.

Exhaust Closes.	
On center............	Slow speed, 100–250 R.P.M.
2° after center.........	Small high speed, 500–800 R.P.M.

The falling of curve for any lag of valve closure can be drawn by laying off (Fig. 140) *gg'*, the retardation, and drawing a gradually rising curve to meet the crest of the regular piston velocity curve *ab''c''*, etc., to *g'*. This crest is above the zero-circle a distance *mn*, Fig. 140, equal to $\frac{\Delta}{4}$ for the lowest possible gas exit velocity, or any distance *mn*, Fig. 141, equal to $h =$ the valve-lift for some constant gas velocity, when this lift is less than $\frac{\Delta}{4}$.

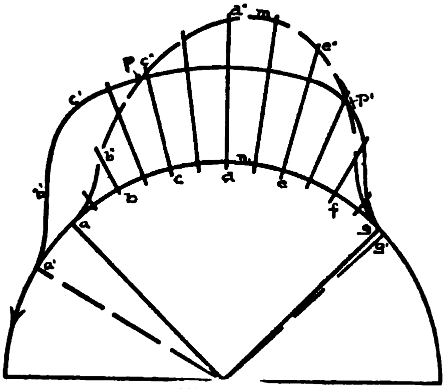


Fig. 140.

The rapid rise and large advance on the rise are best secured by advancing the curve of piston velocities *ab''c''* to *a'b'c'*, which will give as quick a rise as is practicable, and in some cases too quick.

The usual easements at corners give two curves, that of Fig. 140 having a variable velocity of gases throughout, but the least possible where crossing at *P* and *P'* or running on the piston velocity curve of altitude $\frac{\Delta}{4}$.

Any openings beyond this curve are useful on the advance for quickly lowering pressures, and on the drop to account for a little inertia and gas friction. The curve of Fig. 141 will always be an easier curve in rise and fall, and will approach more nearly to constant velocity of exit after the initial opening for equalization of pressure during the first period of exhaust.

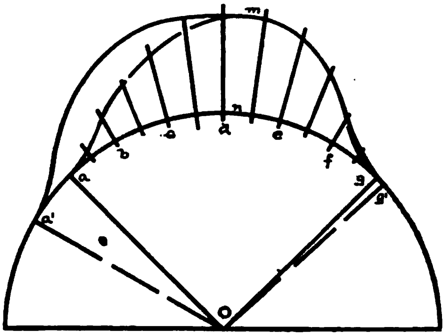


Fig. 141.

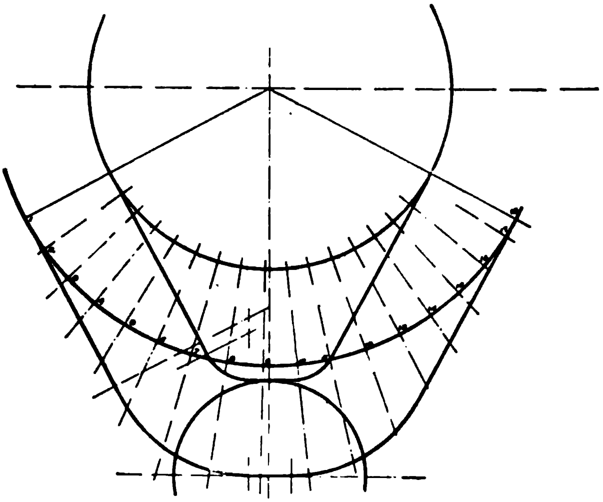


Fig. 142.

Applying the same method of treatment to the common form of exhaust-cam, consisting of two flat faces and one circular connecting arc, shown in Fig. 142, there results the pair of curves of Fig. 143

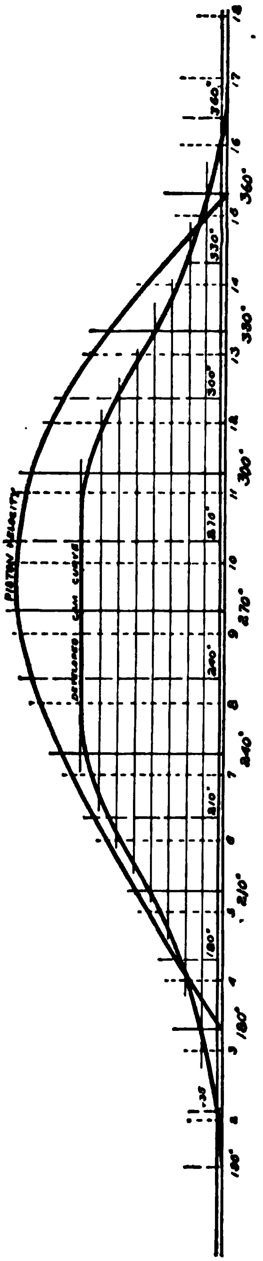


Fig. 143.

These two curves are respectively the piston-velocity curve and the valve-stem motion curve. The interpretation of their relative positions can be easily made from what has already been said.

110. Piston-heads. — Piston-heads are generally flat plates fixed at the edges and uniformly loaded, for which Grashof gives

$$T=\sqrt{\frac{2d^2P_m}{34S_p}}=d\sqrt{\frac{1}{6}\frac{P_m}{S_p}}$$

or

$$S_p=\frac{1}{6}\frac{d^2P_m}{T^2}, \quad . \quad . \quad . \quad . \quad . \quad (188)$$

where d = diameter in inches, which may be taken as the cylinder diameter;

P_m = maximum pressure in the cylinder;

S_p = working stress in the metal.

Small engines usually have flat unstayed heads of thickness varying from

$$t_p'=0.04d \quad . \quad . \quad . \quad . \quad (189)$$

$$\text{to} \quad t_p''=0.08d, \quad . \quad . \quad . \quad . \quad (190)$$

with an average of 0.06d, indicating for maximum pressure of 300 lbs.

S_p = 31,400 when t_p' = 0.04d,
S_p = 7,800 " t_p'' = 0.08d,
S_p = 14,400 " t_p''' = 0.06d,

or for a maximum of 450 lbs.

S_p = 47,000 when t_p' = 0.04d,
S_p = 21,000 " t_p'' = 0.08d,
S_p = 11,800 " t_p''' = 0.06d.

The above stresses are correct only on the theoretic assumption that the real stress is considerably less.

Pistons over 6″ diameter should be web-stayed for stiffness, and preferably be arched outward or inward for a reduction of thickness, which may be $0.6t_p$ for a full arch. Webs are usually about $0.6t_p$ in thickness. These heads run through a great range of thickness, the lower limit holding, but heads are made with very much less stress, as t_p sometimes reaches as high a value as

$$t_p^{\text{IV}} = 0.11d, \quad \ldots\ldots \quad (191)$$

involving stresses of

$S_p = 4200$ for 300 lbs. maximum pressure,
and $S_p = 6250$ " 450 lbs. " "

considered as flat unstayed plates.

The actual stresses are less, but the increased thickness used in large air-cooled pistons is to compensate for the fact that the centers of such heads run very hot.

Double-acting pistons enclosing water-jackets should have heads of thickness about

$$t_p^{\text{V}} = 0.07d \quad \ldots\ldots \quad (192)$$

if unstayed, but when properly stiffened this may be as low as

$$t_p^{\text{VI}} = 0.034d. \quad \ldots\ldots \quad (193)$$

111. **Piston Length.**—There is a rubbing on the sides of the cylinder due to the side thrust of the connecting-rod when inclined to the cylinder axis, and the length of the piston must be such that the pressure on the sides shall be small enough to prevent this wear being excessive.

Excessive wear results in knocking on reversal of direction of stress and in leakage, since the rings will not pack oval cylinders. The weight of the piston in horizontal cylinders will also add a side-bearing stress constantly downward.

This maximum guide pressure P_g in pounds per square inch of piston area is given by the maximum value of

$$P_g = P_m \tan\phi, \quad . \; . \; . \; . \; . \; . \; . \quad (194)$$

where ϕ = connecting-rod angle to axis; and calling θ the crank-angle,

$$l \sin\phi = r \sin\theta,$$

or

$$\phi = \sin^{-1}\left(\frac{r}{l}\sin\theta\right).$$

This angle ϕ is a maximum when $\theta = 90°$ or $\sin\theta = 1$; hence

$$\text{max. } \phi = \sin^{-1}\left(\frac{r}{l}\right). \quad . \; . \; . \; . \; . \; . \; . \quad (195)$$

These maximum tangents of ϕ are easily found by reference to the table.

$\frac{r}{l}$	Tan [ϕ(max.)] for $\theta = 90°$	$\frac{r}{l}$	Tan [ϕ(max.)] for $\theta = 90°$
$\frac{1}{3} = 0.3333$	0.354	$\frac{1}{5.0} = 0.2000$	0.204
$\frac{1}{3.5} = 0.2857$	0.297	$\frac{1}{5.5} = 0.1818$	0.184
$\frac{1}{4.0} = 0.2500$	0.258	$\frac{1}{6.6} = 0.1666$	0.169
$\frac{1}{4.5} = 0.2222$	0.223		

112. Side Thrust from Gas Pressure and Inertia.—When the engine is horizontal single-acting, the thrust directions are as follows, neglecting inertia:

Expansion.	Thrust	downward,	engine	turning	over.
Compression.	"	upward,	"	"	"
Suction.	"	"	"	"	"
Exhaust.	"	"	"	"	"

Adding the effects of inertia, the expansion thrust will remain downward until somewhere the gas pressures become less than the inertia pressures, and this occurs, with delayed ignition, weak mixtures, and in hit-and-miss-governed engines, on every miss-stroke. In such cases the thrust will reverse in direction. Considering inertia, the suction and exhaust thrusts will reverse during the stroke. If the compression be high compared with inertia, reversal may not occur during compression, but in other cases it will.

Double-acting engines and tandem engines, especially when double-acting, are subject to a considerable number of possible variations of thrust in direction, amount, and uniformity, as can be seen by reference to the effective pressure diagrams given for a great variety of cases. The maximum thrust calculated from the double effects of inertia and gas pressure and adding for horizontal cylinders the dead weights of the parts, gives too variable a quantity to set down to a rule, though Güldner does so by neglecting inertia. He says that this is a nearly constant quantity whose average lies between

$$P_g' = .095 P_m \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (196)$$

and

$$P_g'' = .105 P_m. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (197)$$

This is approximately true for full-load cards, proper ignition, and other conditions at their best, and within narrow limits of inertia such as maintain for medium-sized stationary engines, but not for variation of indicator card from the best form, nor for excessive inertia of the heavy double-acting and tandem engines or the small very-high-speed engines used in boats or automobiles. He also gives the admissible normal pressure, disregarding packing-ring groove, as between

$$P_n = 18 \text{ lbs. per sq. in.}, \quad . \quad . \quad . \quad . \quad (198)$$

$$P_n' = 22 \text{ `` `` `` ``} \quad . \quad . \quad . \quad . \quad (199)$$

going as low as

$$P_n'' = 7 \text{ `` `` `` ``} \quad . \quad . \quad . \quad . \quad (200)$$

when excessive length of piston is not troublesome, as in ordinary stationary engines. Moss in his examination of medium-sized engines found the limits of

$$P_n^{\text{III}} = 9.6, \quad \ldots \ldots \quad (201)$$

$$P_n^{\text{IV}} = 4.8. \quad \ldots \ldots \quad (202)$$

with an average of 6.9.

The relation between these various pressures is seen from the following equations:

Maximum force acting on piston-head $= \frac{\pi d^2}{4} P_m$ pounds,

" " " " guide or between cylinder and walls $= \frac{\pi d^2}{4} P_g$.

But $P_g = \tan(\phi \text{ max.}) P_m$.

If now L_p = piston length, the projected area of the rubbing surface receiving $\frac{\pi d^2}{4} P_m \tan(\phi \text{ max.})$ pounds is $d \times L_p$ square inches; hence the normal pressure is

$$P_n = \frac{\pi d^2 P_m \tan(\phi \text{ max.})}{d L_p} \quad \ldots \ldots \quad (203)$$

The best rule for determining piston length is to keep in mind the desirability of reducing P_n to as low a value as possible and keeping it as constant in amount and direction as possible by combination of cylinders and the combination of phases in connection with appropriate inertia. This, in general, calls for as long a piston as possible. As the wrist-pin for uniformity of thrust must be near the center of the piston, the length will be limited by the possibility of the connecting-rod striking the sides of the piston. Short rods then demand short pistons or the expedient of moving the wrist-pin forward, which is worse than too short a piston when carried to excess.

113. Practical Limits of Piston Length.—The average high-speed engine uses piston lengths L_p shorter as a rule than other engines, the practice being from

$$L_p^{\text{I}} = 1.0d. \quad \ldots \ldots \quad (204)$$

$$\text{to} \quad L_p^{\text{II}} = 1.6d. \quad \ldots \ldots \quad (205)$$

Stationary engines when small have lengths between

$$L_p^{III} = 1.4d \quad \ldots \ldots \quad (206)$$

and $$L_p^{IV} = 2.3d, \quad \ldots \ldots \quad (207)$$

while large engines again reduce the limits to

$$L_p^{V} = 1.2d \quad \ldots \ldots \quad (208)$$

and $$L_p^{VI} = 1.7d. \quad \ldots \ldots \quad (209)$$

The figures for guide-thrust given by Moss are based on a mean length of $L_p = 1.5d$ with a maximum pressure $P_m = 300$ lbs., inertia being neglected, and may be used with these limitations.

Double-acting piston-barrels support besides their own weight that of some water; to hold a proper amount of water and allow of the necessary guide-vanes and partitions, a length of 0.6*d* will usually suffice. But two-cycle engines when double-acting must have a piston whose length depends on the port-opening and stroke, since alternately each side must uncover this port at the stroke end only. When engines are double-acting, cross-head guides take up this side-thrust, and as wear can be compensated by adjustment higher sliding pressures are allowed, ranging from 50 to 100 lbs. per sq. in. This design of gas-engine cross-heads involves nothing except provision for feeding water to piston-rods over the design of steam-engine cross-heads and is here omitted.

It should be remembered that when figuring closely on projected area of piston sliding surface, piston-rings must be counted out.

114. **Piston-barrel.**—Piston-barrels have a double duty to perform. First the side-thrust must be distributed over the cylinder surface uniformly, and for this the barrel must have a thickness such as will prevent springing and so transmit from the wrist-pin boss, where the thrust is received, to each part its share of the load.

Between the wrist-pin and the piston-head the barrel must be in compression from the gas pressure, which gas pressure must be transmitted to the bosses without springing of the parts; this is the second duty of the barrel. Inertia of the two ends of the piston causes the barrel to be alternately in compression and tension; this would call for more metal in the barrel near the boss than at a distance. The practice is to allow a certain thickness of metal approximately uniform between boss and head and a thinning out from boss to open end. Where the packing grooves end, the metal is usually thicker than is necessary, to avoid fractures from sharp corners, though in large pistons the thickening is not so marked. Metal only sufficient to resist fracture from the forces acting will make a piston-barrel too light to prevent springing.

When larger than 6 ins., pistons should be webbed to better distribute stress by increased stiffness. These webs should run from the bosses both ways, but should be especially provided between head and boss and should lie in planes through the cylinder axis or radiating from the boss. Where pistons are very large, over 25 ins. diameter, metal can be economized with good effect by casting some webs in planes perpendicular to the cylinder axis, forming inner rings. These webs may have thickness about 0.8 that of the piston-head, but their thickness, height, number, and distribution are items depending on judgment, the guiding principle being a reduction of weight with requisite barrel, thickness, and necessary stiffness. No sharp corners should be permitted.

Pistons, when large enough to require elaborate web systems, will also require jackets for the head, which may be itself considered as one web of the system.

Barrel thicknesses behind the rings are usually the same as in front of the boss and about 0.6 the thickness of flat heads, though in very small engines it may be as low as $0.5t_p$. When tapered in front of the boss the edge should not be thinner than $0.4t_p$.

115. Piston-rings.—Tightness about pistons is secured by cast-iron rings sprung into grooves for all sizes up to 30 ins., beyond which rings may be made in pieces pressed outward by springs.

The diametral piston clearance in the cylinder is a matter that will depend on experience with each style of engine and will vary with the temperature of the piston, its wall thickness and webbing, and on the temperature and thickness of the cylinder walls.

These are things that cannot be predicted, but must be determined for every style and size of engine by trial. A good method is to ease up all bearings, run the engine and measure the time it takes to stop with gas cut-off and no load, engine being hot for first run and cold for the next.

It is certain, however, that the heads of unjacketed pistons will get hotter than other parts, and a preliminary guide to the reduction of diameter toward the head is to reduce the diameter 0.003 in. for every 6 ins. of diameter, beginning at the rings and increasing toward the head, but not uniformly. The number of rings varies from three to ten, and they are best placed one in a groove, though sometimes two in a groove are used. A clearance of about 0.001 in. is allowed at the sides to prevent sticking from heating and gumming of oil.

A piston-ring is a beam when acting as a spring; it therefore has a thickness at the open end less than at the middle. Cast iron in such a beam suffers considerable permanent set under load, but once permanently set for a given load will be almost perfectly elastic for all less loads. It therefore is cut larger than the cylinder and compressed to a diameter less than the cylinder. It should then have acquired all its permanent set and act as a spring; removal may cause a set in the opposite direction, which is of no consequence provided the rings are not too often subject to this reversal of stress.

Original turning diameters may be made 0.175 in. larger than the cylinder diameter before cutting for small rings. For medium-sized rings 8 to 18 ins. this may be 0.25 in. and for very large rings Seaton, in his "Marine Engineering," recommends an increase of $0.01d$ for the original cutting diameter. The widths of the rings W_r in practice are not usually calculated, but made some even number of inches for a certain range of sizes.

It will be found that for small high-speed engines where 13 ins. $< d <$ 6 ins.,

$$W_r = \tfrac{1}{4} \text{ in.} \qquad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (210)$$

This is increased in successive sizes to

$$W_r' = \tfrac{3}{4} \text{ in.} \qquad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (217)$$

in the largest engines, though rings are sometimes met with over an inch wide.

The bridge b between rings serves merely to drag one ring along and may be any width, the best practice making

$$b = 1.0W. \qquad (218)$$

When rings are to be kept close b may be as low as

$$b' = 0.5W, \qquad (219)$$

but never less. With plenty of room, it may be larger, but too much space increases weight needlessly. The maximum is

$$b'' = 1.5W. \qquad (220)$$

If weight must be increased for inertia adjustment this is one good place to do it.

The thickness t_r of rings at greatest cross-section Güldner, who recommends light rings, gives as

$$t_r' = \tfrac{1}{30}d + \tfrac{5}{64}\text{ in.} \qquad (221)$$

A common steam-engine rule given by Kent makes

$$t_r'' = \tfrac{1}{30}d + \tfrac{1}{8}\text{ in.} \qquad (222)$$

It makes very little difference just how this thickness varies, but it should not differ much from the width of ring. The proportions of many good rings can be found in any of the works on machine design, the gas-engine introducing nothing new.

Large sectional rings are best made of antifriction or white metal, and often the end of the piston may also be so lined if efficiently cooled.

116. Wrist-pin.—Single-acting engines carry the wrist-pin in the piston and its size is limited accordingly. When an engine is double-acting, a cross-head is provided to carry the wrist-pin, which may then be as large as desired.

The stresses in the wrist-pin cannot be known without combining the inertia and gas-pressure diagrams, but as a guide, the stresses from gas pressure alone may be used with the limitation that this will set up the highest possible stress, probably at or near dead-center, except when weight of parts or speed are such as make the inertia forces greater than the maximum gas pressure. This latter will occur only in cases of unusual inertia or in cases where inertia for good reasons has been increased above the normal present practice. On starting when speed is low the maximum gas pressure always exceeds any after effective pressure, including inertia.

A wrist-pin is subject to bending and must be strong enough to resist; it also has a tendency to heat, measured by its pressure per square inch of projected area and coefficient of friction.

The temperature will be stable and constant when the rate of heat generation is equal to that of radiation or other means of carrying off. This friction introduces as another though negligible stress some torsion.

To prevent heating of wrist-pins on large marine engines, the pressure is never allowed to exceed 1200 lbs. per sq. in. of projected area. In general, pin sizes great enough for this will be amply strong. It is not possible to increase the length and diameter of the gas-engine wrist-pins without limit. Pins are fixed in the piston-boss, and the length of this boss varies from the maximum $0.25d$ to something less, boss length being sacrificed to give greater length on the bearing part of the pin between bosses.

Let l_p = length of wrist-pin bearing in connecting-rod end;

d_p = diameter of wrist-pin;

F_p = maximum force in pounds on the pin;

$F_p=\frac{\pi d^2}{4}\times P_m$ or $0.00034WN^2r(1+a_1)$, as the case may be;

P_p = bearing pressure per square inch of projected pin-bearing area;

S_b = working stress in the metal subject to bending;

then will the projected area be

$$l_p\times d_p \quad \text{and} \quad P_b=\frac{F_p}{l_pd_p}. \quad . \quad . \quad . \quad . \quad . \qquad (223)$$

For a circular beam supported at the ends and uniformly loaded the following formula expresses the relation between load, fiber, stress, and dimensions:

$$\text{Load}=\frac{\pi}{4}\frac{S_bd_p{}^3}{l_p}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (224)$$

Considering the maximum load as due to gas pressure only, which is not always true, this gives

$$\frac{\pi}{4}\frac{S_b d_p^3}{l_p} = \frac{\pi d^2}{4} P_m;$$

whence

$$S_b = \frac{l_p d^2 P_m}{d_p^3}. \quad \ldots \ldots \ldots \quad (225)$$

117. Limits in Wrist-pin Dimension.—Small engines have pins between the limits:

$$\text{Minimum} \quad d_p = 0.15d; \quad \ldots \ldots \quad (226)$$

$$\text{Maximum} \quad d_p' = 0.25d; \quad \ldots \ldots \quad (227)$$

$$\text{Mean} \quad d_p'' = 0.22d. \quad \ldots \ldots \quad (228)$$

Their bearing length is almost invariably $l_p = .5d$, so that

$$\text{Minimum} \quad l_p d_p = 0.075d^2; \quad \ldots \ldots \quad (229)$$

$$\text{Maximum} \quad l_p d_p' = 0.125d^2; \quad \ldots \ldots \quad (230)$$

$$\text{Mean} \quad l_p d_p'' = 0.110d^2. \quad \ldots \ldots \quad (231)$$

Considering gas pressure along this gives

(1) $P_b = 10.5$ and $P_b = 3150$, when $P_m = 300$,
or $P_b = 4725$, when $P_m = 450$;
(2) $P_b = 6.3$ and $P_b = 1890$, when $P_m = 300$,
or $P_b = 2830$, when $P_m = 450$;
(3) $P_b = 7.15$ and $P_b = 2145$, when $P_m = 300$,
or $P_b = 3217$, when $P_m = 450$.

Considering the stresses from gas pressure alone for the three cases,

(1) $S_b = 148$, $P_m = 44{,}400$, when $P_m = 300$,
or $S = 66{,}600$, when $P_m = 450$;
(2) $S_b = 32$, $P_m = 9{,}600$, when $P_m = 300$,
or $S = 14{,}400$, when $P_m = 450$;
(3) $S_b = 47$, $P_m = 14{,}100$, when $P_m = 300$,
or $S = 21{,}150$, when $P_m = 450$.

For the sort of load these pins receive, nothing but very fine material will stand the higher stresses, even supposing the maximum pressures given were higher than sustained. Mild-steel wrist-pins should not be subjected to more than 15,000 lbs. sq. in. stress or bending, heating, and excessive wear, if not fracture, will result.

Medium-powered and large engines have pins of relatively larger diameter, the limits being

$$\text{Minimum } d_p''' = 0.22d; \qquad (232)$$

$$\text{Maximum } d_p^{\text{IV}} = 0.45d; \qquad (233)$$

$$\text{Mean } d_p^{\text{V}} = 0.31d; \qquad (234)$$

the lengths being about the same as for small engines, giving bearing pressures for these cases of

$$\text{Maximum } P_b = 7.15P_m;$$

$$\text{Minimum } P_b = 3.5P_m;$$

$$\text{Mean } P_b = 5.1P_m.$$

The corresponding stresses for gas pressures acting alone on the pin supposed a beam will be

$$\text{Maximum } S = 47P_m;$$

$$\text{Minimum } S = 11P_m;$$

$$\text{Mean } S = 34P_m.$$

A pin of diameter

$$d_p^{\text{VI}} = 0.35d \qquad (235)$$

should make a good wearing, reasonably stiff pin that will remain well lubricated, especially if by using extra-heavy bosses well webbed the length of the bearing be increased to at least

$$l_p = 0.6d_p. \qquad (236)$$

118. Connecting-rods.—When ignition is early and all other conditions best the highest cylinder pressures occur at the beginning of the stroke. Should this not happen the highest pressures resulting will never be as great as for the former case. The thrust on the connecting-rod is greater than the piston load, and if the cylinder pressure were always constant, the maximum thrust would occur somewhere after the beginning of the stroke. It is therefore necessary in steam-engine work to find the maximum angle of the rod to get the maximum thrust, but in gas-engine work the maximum thrust possible occurs at dead-center when the mixture and ignition are best and weight of charge greatest. The maximum possible cylinder pressure times the piston area will then give the maximum possible thrust on the rod, when the inertia is less than this. Hence to get the real maximum thrust, inertia and gas pressure must be known for the beginning of the stroke.

The rod is a column, as its length is always great enough to cause it to fail by bending rather than by crushing when under compression. Rods for single-acting cylinders are always in compression with respect to gas pressures, but in tension for inertia at certain parts of the stroke. When double-acting and tandem cylinders act through one rod the different parts of the rod are alternately in tension and compression for gas-pressure stress. Inertia of the rod itself introduces bending stresses in the rod which it must resist as a beam.

Thus a connecting-rod must suffer compression, flexure, tension, and must be strong enough to resist all of these. The maximum pressures in gas-engine cylinders when slow-speed and the high inertia for even the small high-speed types call for rods large enough to be able to resist the bending due to rod inertia generally. But when careful design for minimum weight is needed the complete analysis should be made.

There is nothing except the determination of the maximum thrust and tension and their possible dependence on initial inertia, rather than gas pressure, in the design of gas-engine rods over what has long been practiced for steam-engine rods. This will probably affect the constants of those formulas, so this point will be examined.

For steam-engine rods, formulas for the diameter at the middle have been used in two forms, one giving the rod as a function of square root of cylinder diameter, rod length l_r, and fourth root of the pressure

$$d_r=a_r\sqrt{dl_r\sqrt{P_m'}}. \qquad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (237)$$

Whitham's, Marks's, and Thurston's formulas are of this form, but Thurston found it advisable to add a term dependent on speed.

In this formula P_m' is the maximum pressure at the piston at the beginning of the stroke, whether due to inertia of reciprocating parts or to gas pressure. The other form of formula drops the length term and becomes Sennett's form,

$$d = b_r d\sqrt{P_m'}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (238)$$

in which the value of d_r is given for long and for short rods differently. We have then

$$d_r = a_r\sqrt{dl_r\sqrt{P_m'}} + C_r \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (239)$$

$$\text{or} \quad d_r = b_r d\sqrt{P_m'}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (240)$$

in which

$a_r = 0.0272$, $C_r = 0$ (Whitham);
$a_r = 0.02758$, $C_r = 0$ if $d_r < \frac{l}{24}$ by his other form (Marks);
$a_r = 0.15$, $C_r = 0.5''$ for fast engines (Thurston);
$a_r = 0.08$, $C_r = 0.75''$ for slow engines (Thurston);
$b_r = 0.01818$ (Sennet);
$b_r = 0.0179$, when $d_r > \frac{l}{24}$ (Marks);
$b_r = 0.0229$ for short-stroke engines (Seaton);
$b_r = 0.0241$ for long-stroke engines (Seaton);
$b_r = 0.021$ as an average of the above (Kent).

The above is for round rods. When rectangular, of thickness t and depth d, and substituting t for d_r in above,

$$a_p = 0.0209, \quad C_p = 0.47 \text{ (Thurston)},$$

the depth d at crank is $2\frac{1}{4}t$ and at cross-head $1\frac{1}{2}t$.

119. Limiting Dimensions for Connecting-rods.—Small round rods for automobile-engines usually have mid-section diameters between

$$d_r = 0.011 d\sqrt{P_m'} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (241)$$

$$\text{and} \quad d_r = 0.014 d\sqrt{P_m'}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (242)$$

and these are for lengths varying from 11 to 14 diameters. This makes a rod lighter than results from any of the steam-engine formulas. Gas pressures seldom exceed 250 lbs. per square inch in these engines.

Plain rectangular rods are more common than round ones in these engines, and for these the average thickness is

$$t_r = 0.008d\sqrt{P_m'} \quad . \ . \ . \ . \ . \ . \quad (243)$$

for rods varying from 15 to 25 thicknesses in length. The width at the crank end is usually $2.3t$ and at the piston end $1.6t_r$. Sometimes an I-beam section is used—in which case the breadth of the top flange is $1.3t_r$ for a flat rod, while the channel cuts away approximately one fourth of the thickness on each side.

Medium-powered stationary engines with piston 6″ to 15″ use rods between the limits of

$$\left.\begin{aligned} d_r &= 0.025\sqrt{dl_r\sqrt{P_m'}} \\ \text{and}\quad d_r' &= 0.031\sqrt{dl_r\sqrt{P_m'}} \end{aligned}\right\} \quad . \ . \ . \ . \ . \quad (244)$$

for rods varying in length from 12 to 20 diameters, showing a fair accord with steam practice as represented by the Whitham and Marks formulas.

Very large engines have rods based on these formulas almost without exception and are generally round in section.

Piston-rods, stuffing-boxes, and cross-heads are designed precisely as for steam-engines, the only variation being in the determination of the maximum stresses by the effect of combining inertia and gas-pressure curves for the proper weights, speeds, combination of cylinders, and phases of the cycle.

120. **Crank-shafts, Pins, and Cranks.**—Gas-engine cranks, shafts, and pins, while designed by the same general method as are the parts of the steam-engines, will have different proportions expressed in terms of power output because of the greater variety of stresses met with. The maximum stress for a single-throw crank-shaft varies more and more irregularly from the mean with conditions, than does the same relation for steam, while for multiple-throw shafts

the variation of the relation and extent of the maximum does not vary with the number of cranks at all like the similar cases with steam. The reason is to be found in the greater variety of phases per cycle, and the phase relations in combining cylinders, and the greater number of different cylinder combinations practiced. It is not necessary to investigate the variations of stress with load ignition and other similar conditions for the design of crank-shafts, but variations due to multiplying cranks, when acted on by full-load gas pressures and by the various inertia stresses for the maximum speeds and weights of reciprocating parts, must be taken into consideration. The resulting crank-shaft system may especially, when of multiple-throw, seem out of all proportion to similarly numbered steam crank-shafts.

Power-cylinders of gas-engines nearly always act on center cranks, though occasionally overhung cranks are used to drive the suction-cylinders for two-cycle engines. Shafts with pins and crank-arm are usually made in one piece, but the stresses in the various parts are different in intensity and nature.

Forces transmitted through the connecting-rod are due to the combined action of gas pressure and inertia of reciprocating parts, and have the effects of bending the crank-pin which resists as a simple beam, bending the crank resisting as a cantilever beam, and bending and twisting the crank-shaft. Of course there are various shears, but need not be separately considered. With two or more cranks the twisting of the shaft due to the connecting-rod forces on the forward crank-pin, is transmitted through the forward crank-arm of the after crank to forward end of the after crank-pin, causing another bending force and moment on the after crank-pin. Dead loads, such as the weight of fly-wheels, dynamo-armatures, belt-pulls, and pulley-weights, all act on the shaft to bend it, and constitute another set of active forces that must be resisted.

A complete discussion of all the forces excepting the last set acting on a crank-shaft, and the complete process in detail leading to the design of such a shaft, is given by Prof. Dunkerly in a paper before the Institution of Naval Architects of Great Britain, and may be referred to with profit; it is reprinted April 4 and 11, 1902, in *London Engineering*. Professor Kline, in the "High-speed Steam-engine," gives a simple, graphic method for determination of the stresses and their resultant moments. For the purposes of this work the method will be merely outlined.

121. Maximum Twisting Moment.—Consider first the shaft to resist the twisting effort. The maximum twisting moment on the shaft is given by the pressures on the turning-effort diagram. This must be constructed, since, when gas pressures are a maximum, inertia has the greatest effect in reducing the effective force, and at this time the moment arm is zero. When, by rotation, the arm increases, the effective pressures may increase or decrease, depending on the inertia, and assuming always full-load indicator cards, so that prediction of maximum moment is difficult.

For approximate work the results of measurements on other turning-effort diagrams may be used; these results give the ratio R_m of maximum turning moment to mean turning moment, and of course the mean turning moment is easily found from the indicated horse-power of the engine.

Let M_t' = mean turning moment in inch-pounds;

M_t = maximum " " " "

$$R_m = \frac{M_t}{M_t'} = \frac{\text{maximum turning force}}{\text{mean turning force}}; \quad \ldots \ldots \quad (245)$$

then mean turning moment in foot-pounds is

$$\frac{\text{I.H.P.} \times 33000}{2\pi r N} \times r.$$

$$M_t' = \frac{\text{I.H.P.} \times 33000}{2\pi N} \times 12 \text{ inch-pounds.} \quad . \quad . \quad (246)$$

$$= 63025 \frac{\text{I.H.P.}}{N} \text{ inch-pounds.}$$

$$M_t = 63025 \frac{\text{I.H.P.}}{N} R_m. \quad \ldots \ldots \ldots \quad (247)$$

An examination of turning-effort diagrams with different inertia conditions and cylinder combinations has given the following values for R_m, which are true only for the conditions given, but may be approximately true for nearly similar cases. The principal conditions are:

(*a*) Cylinder combination to produce a certain angle between phases and cranks.

A complete cycle is completed in 720° of crank movement; hence if expansion begins in number two cylinder 120° of crank-angle after it begins in number one cylinder, the phase difference is 120°.

(*b*) The original form of indicator card, which, of course, is always taken for full load.

(*c*) The inertia, which is measured by four quantities, N, $\frac{W}{a}$, r, and $\frac{r}{l}$; but as two of these enter as factors of a product $N^2\frac{W}{a}$, this product is best given as the determining condition; or to be more complete, the factor r may be properly included in the product giving $N^2\frac{W}{a}r$. This quantity may be the same for many engines, but always gives the same inertia forces for the reciprocating parts.

It should be noted that beginning with low inertia the ratio R_m decreases with increase of inertia because this lowers effective pressure for the first half stroke, while a high inertia approaching the initial gas pressure so raises terminal effective pressure as to increase the ratio. With inertias greater than the gas pressure the numerator of the fraction would be due to this alone. For a single-cylinder engine the mean turning effort is constant for full load and changes in the R_m are due to changes of the maximum only. This mean increases directly with the number of cylinders, while the maximum varies a little both ways.

Reciprocating parts may be combined in sets to reduce the resultant inertia forces that, where present, often so distort the turning-effort diagram as to give erratic R_m values. For two-cycle engines approximate values can be obtained by multiplying R_m for four-cycle by 1.7. This presupposes a reduced indicator card of 0.6, the mean pressure.

122. Shaft to Resist Twisting Only.—The resisting moment of a shaft subject to torsion only is

$$\frac{\pi}{16}d_s^3S_s = 0.1963d_s^3S_s;$$

VALUES OF $R_m = \frac{\text{MAXIMUM TURNING MOMENT}}{\text{MEAN TURNING MOMENT}}$.

CASE.	R_m.	Inertia Conditions Expressed by the Value of $N^2\frac{W}{a}r$.	Maximum Inertia Pressure, Approximate.	Indicator Card.	$\frac{r}{l}$	Cylinder Conditions (All Single-acting, Four-cycle.
I	10.0	61,685	25	Carb. water-gas. Compression 65 pounds above atmosphere; maximum pressure 240 pounds above atmosphere; ignition slightly late but correct	¼	Single cylinder, four-cycle
	9.2	123,370	50	do.	"	" " " "
	7.4	246,740	100	do.	"	" " " "
II	9.0	37,500	15	Same with earlier igniton	"	" " " "
	6.4	204,200	85	" " " "	"	" " " "
	9.6	504,000	210	" " " "	"	" " " "
III	10.0	37,500	15	Produce gas with compression 90 pounds and maximum pressure 180 pounds above atmosphere; ignition correct	"	" " " "
	8.2	204,200	85			
	14.0	504,000	111			
IV	4.5	61 685	25	Same gas as Set I	"	Two cylinders { phase difference 360° cranks 2
	3.5	"	"	"	"	Three cylinders { phase difference 240° cranks 3
	1.7	"	"	"	"	Four cylinders { phase difference 180° cranks 4
	2.15	"	"	"	"	Five cylinders { phase difference 144° cranks 5
	1.6	"	"	"	"	Six cylinders { phase difference 120° cranks 6
	1.5	"	"	"	"	Eight cylinders { phase difference 90° cranks 8
V	3.5	123,370	50	Same as Set I	"	Two } Phases and cranks as for Set IV
	3.2	"	"	"	"	Three
	1.8	"	"	"	"	Four
	2.1	"	"	"	"	Five
	2.1	"	"	"	"	Six
	1.35	"	"	"	"	Eight

VI	4.6	246,740	100	Same as Set I	⅛	Two	Phases and cranks as for Set IV
	3.2	"	"	"	"	Three	
	3.3	"	"	"	"	Four	
	2.15	"	"	"	"	Five	
	1.6	"	"	"	"	Six	
	1.4	"	"	"	"	Eight	
VII	4.2	37,500	15	Same as Set II	⅛	Two	Phases and cranks as for Set IV
	3.1	"	"	"	"	Three	
	1.8	"	"	"	"	Four	
	1.0	"	"	"	"	Five	
	1.5	"	"	"	"	Six	
	1.25	"	"	"	"	Eight	
VIII	4.2	204,200	85	Same as Set II	⅛	Two	Phases and cranks as for Set IV
	3.0	"	"	"	"	Three	
	3.1	"	"	"	"	Four	
	1.8	"	"	"	"	Five	
	1.5	"	"	"	"	Six	
	1.3	"	"	"	"	Eight	
IX	8.0	504,000	210	Same as Set II	⅛	Two	Phases and cranks as for Set IV
	3.7	"	"	"	"	Three	
	7.2	"	"	"	"	Four	
	1.8	"	"	"	"	Five	
	2.5	"	"	"	"	Six	
	1.25	"	"	"	"	Eight	
X	4.5	37,500	15	Same as Set III	⅛	Two	Phases and cranks as for Set IV
	3.65	"	"	"	"	Three	
	1.7	"	"	"	"	Four	
	2.1	"	"	"	"	Five	
	1.6	"	"	"	"	Six	
	1.45	"	"	"	"	Eight	
XI	6.3	204,200	85	Same as Set III	⅛	Two	Phases and cranks as for Set IV
	3.3	"	"	"	"	Three	
	5.0	"	"	"	"	Four	
	2.1	"	"	"	"	Five	
	1.8	"	"	"	"	Six	
	1.5	"	"	"	"	Eight	
XII	13.0	504,000	210	Same as Set III	⅛	Two	Phases and cranks as for Set IV
	5.5	"	"	"	"	Three	
	12.0	"	"	"	"	Four	
	2.0	"	"	"	"	Five	
	4.1	"	"	"	"	Six	
	1.8	"	"	"	"	Eight	

where d_s = diameter of shaft in inches;
S_s = shearing stress in metal.

Equating this to maximum twisting moment and solving for d_s,

$$d_s=\sqrt[3]{\frac{5.1}{S_s}\times 63025\frac{\text{I.H.P.}}{N}R_m}$$

$$=\sqrt[3]{321427\frac{\text{I.H.P.}}{N}R_m}=C_s\sqrt[3]{\frac{\text{I.H.P.}}{N}R_m}. \quad . \quad . \quad (248)$$

For steel shafts S should not exceed 12,000 lbs. This is the shaft diameter that will transmit the power developed if the shaft is subject to no bending. Shafts are always subjected to bending from their own weight, from belt-pulls and fly-wheel lights and from crank-pin forces. This bending is always accompanied by twisting, and the shaft must be strong enough to resist both stresses acting together, and when so built will be strong enough to resist all shears as well. This is done by finding a shaft diameter that will resist a moment of applied stresses equal to the combined turning and bending moments.

123. **Maximum Bending and Twisting Moment.**—Let M_b = maximum bending moment; M_e = equivalent twisting moment;

then

$$M_e=M_b+\sqrt{M_b^2+M_t^2}; \quad . \quad . \quad . \quad . \quad . \quad (249)$$

and for this the shaft must have a diameter in inches, when moments are in inch-pounds, of

$$d_s=\sqrt[3]{\frac{5.1}{S_s}M_e}\text{ inches.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (250)$$

If bending of the shaft between journals, due to load at the crank-pin, be imagined applied uniformly to a continuous shaft l_s inches between journals, the maximum bending moment is

$$M_b=\frac{\text{Maximum applied force}\times l_s}{8}\text{ inch-pounds.} \quad . \quad (251)$$

This is only one of many possible assumptions on the constraining action of a journal, but it is common and simple and seems to work well in practice. The maximum applied force is that maximum force acting on the wrist-pin and may be due to inertia alone in the idle strokes when inertia is higher than gas pressure, or may be due to gas pressure alone when an engine is starting and inertia is zero, or may finally be due to the combination of gas and inertia pressures acting on the piston.

Gas and inertia pressures are additive only at the end of a working-stroke, where terminal gas pressure is added to the lower value of the out dead-centre inertia. Should inertia so increase as to make this sum equal to or greater than the initial gas pressure, the in centre inertia will be greater still for the suction- and exhaust-strokes, so that it will be found that the maximum pressure acting is either initial gas pressure or in center inertia, as the case may be. Inertia pressures exceed gas pressures at inner dead-center only for extraordinarily high speeds or heavy parts with long strokes, which is not found in ordinary gas-engines.

This may be calculated for a given case, but may be taken for medium-powered, medium-speed engines at 300 lbs. per sq. inch. For small high-speed automobile engines 250 lbs. and increased to 450 lbs. as a maximum limit for the largest engines on rich fuel and with the high compression permitted by water-cooled pistons and valves if the speeds do not exceed 250 R.P.M. Whatever it may prove to be, call this maximum pressure at the wrist-pin P'_m lbs. per sq. inch, then

$$M_b=\frac{\pi d^2}{4}\times P'_m\times\frac{l_s}{8}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (252)$$

Putting length between bearing to cylinder diameter $\frac{l_s}{d}=b$, M_b can be found.

This b will average for any number of cylinders up to four on one *crank* the value $b=1.3$. For the average value of b given, the bending moment is

$$M_b=0.128P'_m d^3. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (252)$$

The expression for twisting moment and equivalent moments can be similarly expressed as a function of the cube of the cylinder diameter:

$$M_t = R_m M_t' = R_m \frac{6 \times 33000}{\pi} \frac{\text{I.H.P.}}{N}. \quad . \quad . \quad . \quad (253)$$

Expressing indicated horse-power in terms of cylinder diameter *d*, in number of single-acting 4-cycle cylinders η, and the stroke in inches *l*:

$$\text{I.H.P.} = \frac{\pi d^2 \times P \times l \times N \eta}{4 \times 33000 \times 12 \times 2} = \frac{\pi d^2 P l N \eta}{4 \times 12 \times 2 \times 33000};$$

hence

$$M_t = \frac{d^2 P l \eta}{16} R_m \text{ inch-pounds.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (254)$$

Strokes are always kept within narrow limits in their relation to cylinder diameter. Put the ratio

$$\frac{l}{d} = x, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (255)$$

then

$$M_t = \frac{d^3 P x \eta}{16} R_m. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (256)$$

Hence for twisting stresses only

$$d_s = d \sqrt[3]{\frac{5.1}{S_s} \frac{P x \eta R_m}{16}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (257)$$

To account for bending as well as twisting, instead of M_t the moment M_e is to be substituted on the M_t moment multiplied by the ratio

$$\frac{M_e}{M_t} = \frac{M_b}{M_t} + \sqrt{\left(\frac{M_b}{M_t}\right)^2 + 1}.$$

From the above

$$\frac{M_b}{M_t}=\frac{\pi d^3 P'_m b}{32}\times\frac{16}{d^3 P x \eta R_m}=\frac{\pi P'_m b}{2 P x \eta R_m},$$

whence

$$\frac{M_e}{M_t}=\frac{\pi P'_m b}{2 P x \eta R_m}+\sqrt{\left(\frac{\pi P'_m b}{2 P x \eta R_m}\right)^2+1}. \quad . \quad . \quad . \qquad (258)$$

This gives for the shaft diameter, to resist bend and twist due to live loads at crank-pins and received from the connecting-rod

$$d_s=d\sqrt[3]{\frac{5.1}{S_s}\,\frac{P x \eta R_m}{16}\left(\frac{\pi P'_m b}{2 P x \eta R_m}+\sqrt{\left(\frac{\pi P'_m b}{2 P x \eta R_m}\right)^2+1}\right)}. \qquad (259)$$

It will be well to repeat the meaning of the symbols:

d = cylinder diameter in inches.
S_s = maximum allowable metal stress = 12,000 lbs. for steel.
P = mean effective pressure in the cylinder.
x = ratio of stroke and diameter of cylinder.
η = number of four-cycle single-acting cylinders contributing to the stress of the shaft.
R_m = ratio of maximum to mean turning force.
b = ratio of length between journals to cylinder diameter = length of crank-pin plus twice the thickness of crank-arm to cylinder diameter.

Though there is such a great variety of cases, each with its own particular value for the above expression as to make a mean value impossible or worthless, it is possible to state within what limits the ratio $\frac{d_s}{d}$ does lie in the engines already built. This knowledge will serve as a check on the computation by the formula.

One significant fact is to be noted: the product ηR_m will be nearly constant for any number of cylinders up to four and will vary but little for any number up to eight, all supposed four-cycle single-acting.

Empirical expressions for shaft diameters will also give a fair idea of practice in the use of factors of safety.

124. Limits of Shaft Diameters.—Small single-cylinder high-speed automobile engines have shaft diameters between the limits

$$d_s' = 0.21d \quad \dots \dots \quad (260)$$

and $$d_s'' = 0.34d, \quad \dots \dots \quad (261)$$

with a mean of $$d_s''' = 0.30d. \quad \dots \dots \quad (262)$$

These engines carry rather light fly-wheels, but with the heavier fly-wheels and lower inertia of the larger medium-powered stationary engines the ratio of diameters does not depart much from these limits, as will be seen from the following for this class:

$$\text{Minimum } d_s^{\text{IV}} = 0.28d; \quad \dots \dots \quad (263)$$

$$\text{Maximum } d_s^{\text{V}} = 0.42d; \quad \dots \dots \quad (264)$$

$$\text{Mean } d_s^{\text{VI}} = 0.34d. \quad \dots \dots \quad (265)$$

Large engines in which the impulses of not more than four single-acting four-cycle cylinders, two four-cycle double-acting cylinders, two single-acting two-cycle cylinders or one double-acting two-cycle cylinder are transmitted through one shaft and carrying the usual fly-wheels will have diameters between the limits

$$d_s^{\text{VII}} = 0.40d \quad \dots \dots \quad (266)$$

and $$d_s^{\text{VIII}} = 0.50d. \quad \dots \dots \quad (267)$$

Practically the same limits exist for less cylinders, always supposing no two to work in phase.

Heavy belt loads, fly-wheels, or dynamo armatures are best carried between one main journal and an outboard bearing, which does not throw any of the bend on the crank-shaft proper and permits the use of the simpler formulas for equivalent moments for this shaft, while the thickening of the shaft where the dead weight comes reduces the deflection.

All the dead loads must be resolved into a resultant, and if they are in one plane the resultant is found by the parallelogram of forces.

The thickness of the shaft at the place where the concentrated resultant rests will be determined by the allowable deflection rather than by strength considerations.

For such work a convenient formula is that of Mr. W. H. Booth, given in "American Machinist":

$$d_s^{\text{IX}} = 3\sqrt[3]{\frac{W_w D_j}{2}}, \quad \ldots \ldots \ldots \quad (268)$$

where d_s^{IX} = diameter at the load;
W_w = load in tons of 2240 lbs.;
D_j = distance in feet to center of journals.

This will give a deflection of

$$h_s = \frac{P_w \times d_j^3}{48EI} \text{ inches.} \quad \ldots \ldots \ldots \quad (269)$$

In the above

P_w = load in pounds;
d_j = span in inches;
E = modulus of elasticity = 30,000,000 for steel;
I = moment of inertia = $0.0491(d_s^{\text{IX}})^4$.

Any deflection gives a slope to the axis, and this slope or tangent of angle of deflection is

$$R_s = \frac{W_w D_j^2}{70 d_s'}. \quad \ldots \ldots \ldots \quad (270)$$

124. Main Bearings.—All forces acting on the crank-pin cause reaction on the bearings and produce between the shaft and bearing a variable pressure. To keep oil from squeezing out the maximum pressure should not exceed a certain amount, nor should the mean exceed a certain amount. If the mean pressure is low a sudden rise to quite a considerable maximum is not harmful. Vertical engines usually have shorter bearings between cylinders than at the ends of the crank-shaft. These bearings are equidistant from the center-line of the rod. In cases like this the length of the inner bearing is made about equal to the shaft diameter, while the outside bearings are about two or two and one-half shaft diameters in length.

Let l_j = the sum of the length of two adjacent journals carrying load from one cylinder;
P_j' = maximum journal pressure on projected area;
P_j = mean journal pressure on projected area;
η = number of single-acting four-cycle cylinders acting on one pair of journals;
P_m' = maximum cylinder pressure;
P = mean cylinder pressure for each working-stroke;
then $\frac{P}{4}$ = " " " " cycle of one cylinder;
$\frac{\eta P}{4}$ = " " " " " " η cylinders.

$$\frac{\pi d^2}{4}\,\frac{\eta P}{4} = P_j l_j d_s,$$

$$\text{or} \quad P_j = 0.098\eta P \frac{d^2}{l_j d_s}. \qquad (271)$$

125. Journal-pressure Limits.—For large multicylinder and double-acting engines the mean journal pressure will run from

$$P_j' = 0.3P \qquad (272)$$

$$\text{to} \quad P_j'' = 0.5P \text{ lbs. sq. in.} \qquad (273)$$

Medium-powered single-cylinder engines are subject to lower mean pressure for the cycle, though a little higher for the one working-stroke, the limits being

$$P_j''' = 0.13P \qquad (274)$$

$$\text{and} \quad P_j^{\text{IV}} = 0.16P, \qquad (275)$$

while for automobile engines about the same average maintains:

$$P_j^{\mathrm{v}}=0.17P. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (276)$$

Maximum pressures are sometimes quite high:

$$\frac{\pi d^2}{4}\times P_m{}'=P_j{}'l_jd_s;$$

$$\therefore\ P_j{}'=\frac{\pi}{4}\frac{d^2}{l_jd_s}=0.39\frac{d^2}{l_jd_s}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (277)$$

These maxima lie between

$P_j{}'$.		
Maximum.	Minimum.	Engine.
$1.20P_m{}'$	$0.65P_m{}'$	Automobile engine
$1.30P_m{}'$	$1.65P_m{}'$	Stationary medium size.

The average for large engines will be about $1.25P_m{}'$. It should be remembered these maxima $P_m{}'$ are different, depending on conditions given before, and that ordinarily they exist for only a short time for single cylinders, the period increasing on increase of number of cylinders.

126. Crank-pins.—A crank-pin is a beam and may receive two principal loads, each causing a bending stress. First, the after crank-pin or that nearest the load receiving the stress from a crank farther away will be subject to a bend from a force applied at the end farther from the load whose maximum amount is the maximum rod-thrust or pull. Secondly, it receives uniformly over its length the force due to its own rods, maximum thrust or pull. Steam-engines when multiple-cylindered usually have the work equally divided and applied at all cranks simultaneously. In such cases the transmitted bending force will be ½, ⅓, etc., according to the number of cylinders times the total turning force shown by the combined turning-effort diagram. Four-cycle gas-engines work one stroke in four; hence, so far as the gas-pressure forces are concerned, when cranks are parallel or at 180° or 360° the maximum force transmitted to the forward end of the after crank-pin will be simply the maximum turning force due to one cylinder, which is the same as that for any number of cylinders on separate cranks up to four. The same is true for two-cycle cylinders, with cranks at 180°, up to two in number. This does not, however, apply to turning effort due to inertia forces, which also affect turning efforts. The resultant inertia forces for all forward cranks together will be transmitted to the end of the after crank-pin.

A two-cylinder four-cycle engine with two cranks will give to the end of the forward crank a maximum force equal to the maximum turning force for one cylinder; similarly for three cylinders and three cranks the after pin receives at its forward end the maximum turning force for two cylinders. The ratio R_m of maximum turning force for $\eta-1$ cylinders to the mean turning force for $(\eta-1)$ cylinders is given for various cases. Dividing this by $\frac{\eta}{\eta-1}$ gives the ratio of maximum for $(\eta=1)$ to the mean for η cylinders, call the result R_m'. Then the maximum force F_{cp} acting on the forward end of the after crank-pin is

$$F_{cp}=63{,}025\frac{\text{I.H.P.}}{rN}R_m', \quad . \quad . \quad . \quad . \quad . \quad (278)$$

where I.H.P. is the indicated horse-power for n cylinders.

The bending moment for this force is

$$B_{cp}=F_{cp}l_{cp} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (279)$$

inch-pounds, when l_{cp} = length of crank-pin in inches.

Uniformly distributed over the pin is a force whose maximum is probably the maximum effective piston pressure for one cylinder P_m' times the piston area, and the bending moment for this is

$$B_{cp}'=\tfrac{1}{8}\left(\frac{\pi d^2}{4}P_m'\times l_{cp}\right). \quad . \quad . \quad . \quad . \quad . \quad (280)$$

These two forces are not a maximum at one time, nor do they act along the same lines nor produce bending in the same plane. The worst case that could occur, however, or rather a worse case than could ever occur, would be to have them act together to produce additive effects. For added moments the diameter of pin necessary would be given by

$$\frac{d_{cp}{}^3 S_{cp}}{5.1} = \left(63025\frac{\text{I.H.P.}}{rN}R_m{}' + \frac{\pi d^2 P_m{}'}{32}\right) l_{cp};$$

$$\therefore \quad \frac{d_{cp}{}^2 S_{cp}}{5.1} = \left(63025\frac{\text{I.H.P.}}{rN}R_m{}' + \frac{\pi d^2 P_m{}'}{32}\right)\frac{l_{cp}}{d_{cp}}.$$

Hence

$$d_{cp} = \sqrt{\frac{S_{cp}}{S_{cp}}\left(63025\frac{\text{I.H.P.}}{rN}R_m{}' + \frac{\pi d^2 P_m{}'}{32}\right)\frac{l_{cp}}{d_{cp}}}, \quad . \quad . \quad (281)$$

$$d_{cp} = d\sqrt{\left(\frac{PR_m{}'}{\times 16} + \frac{\pi P_m{}'}{32}\frac{l_{cp}}{d_{cp}}\right)\frac{5.1}{S_{cp}}}. \quad . \quad . \quad . \quad . \quad (282)$$

Gas-engine practice sanctions pins in which the ratio of length to diameter is as follows:

$\frac{l_{cp}}{d_{cp}} = \frac{\text{length of crank-pin}}{\text{diameter of crank-pin}}$.		
Maximum.	**Minimum.**	**Mean.**
1.7	0.90	1.4

Diameters calculated by the above should lie between the following limits for strength:

d_{cp} = diameter of crank-pin.			
Maximum.	**Minimum.**	**Mean.**	**Engine.**
$d_{cp}=0.20d$	$d_{cp}=0.38d$	$d_{cp}=0.31d$	Automobile
$d_{cp}=0.31d$	$d_{cp}=0.47d$	$d_{cp}=0.40d$	Medium stationary
$d_{cp}=0.35d$	$d_{cp}=0.52d$	$d_{cp}=0.47d$	Large stationary

Strength is not the only question of importance: sufficient projected surface for proper lubrication under the maximum pressures is necessary on the assumption of equal distribution of stresses over the surface.

Let P_{cp} = maximum pressure on crank-pin projected area;

then
$$P_{cp}=\frac{\frac{\pi d^2}{4}P_m'}{l_{cp}\times d_{cp}}=0.785\left(\frac{d^2}{l_{cp}\times d_{cp}}\right)P_m'. \quad . \quad . \quad . \quad (283)$$

The values of this ratio practised are given below and are purely empirical, hence are not to be considered as limiting future designs.

$\frac{d^2}{l_{cp}\times d_{cp}}$.		
Maximum.	Minimum.	Mean.
10.0	4.0	5.5
5.5	3.1	4.0

Most formulas for crank-pins to prevent overheating depend on an assumed coefficient of friction with a mean journal pressure depending on I.H.P. and velocity of rubbing surface, but as experience permits the former empirical treatment it seems safer. It should be noted in adjusting the crank-pin surface that length is more effective than diameter in cooling, and of two pins otherwise equal in all respects, including projected areas, the longer will run cooler, though weaker for the same applied forces.

127. Cranks.—Cranks are cantilever beams with moment arm r, of thickness t_c parallel to shaft and width w_c parallel to the plane of rotation, are always rectangular in section and, except in very large sizes, of uniform cross-section. The minimum width may be arbitrarily fixed, so as to take up shaft end-thrust and prevent side play of the crank-shaft, as

$$w_c'=d_s+\tfrac{1}{2}'' \text{ for smallest engines,} \quad . \quad . \quad . \quad (284)$$

$$w_c''=d_s+2'' \text{ for largest engines.} \quad . \quad . \quad . \quad . \quad (285)$$

In small and medium engines the width will lie between

$$1.3d_s < w_c < 1.7d_s,$$

while in large engines this ratio of width to shaft diameter is as low as 1.1.

When power generated in several cylinders is simultaneously applied to several cranks and transmitted through the last or after arm to the driven shaft, this last arm receives a force at the crank radius whose maximum is the maximum turning force for all cylinders. This force is not necessarily greater than for one cylinder, as has been pointed out, and as its moment is the turning moment, the ratios of maximum to mean turning moment given can be used to find the maximum moment on the crank. This gives by equating moments of resistance to moments of load, in inch-pounds,

$$63025\frac{\text{I.H.P.}}{N}R_m = \frac{t_c w_c^2 S_c}{6}, \quad . \quad . \quad . \quad . \quad . \quad (286)$$

or

$$\frac{33000 \times 6}{\pi} \times \frac{P \times l \times \pi d^2 \eta R_m}{2124 \times 33000} = \frac{t_c w_c^2 S_c}{6},$$

for η cylinders, single-acting four-cycle, or

$$\frac{Pld^2 \eta R_m}{16} = \frac{t_c w_c^2 S_c}{6} = \frac{w_c a_c S_c}{6},$$

if a_c = area of cross-section of crank-arm.

This gives

$$a_c = \frac{3}{8}\frac{Pld^2 \eta R_m}{w_c S_c}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (287)$$

or

$$t_c = \frac{3}{8} w_c \frac{Pld \eta R_m}{S} \frac{d}{w_c}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (288)$$

The ratio $\frac{d}{w_c}$ lies between 0.35 and 0.55 for automobile engines up to four cranks, the larger value being somewhat the better. For medium-powered engines it will lie between 0.40 and 0.60, with the average for these as well as for large engines at 0.55. The resulting thickness t_c can be given as a function chiefly of the width for an engine of a given size and class and when calculated will lie between

$$t_c' = 0.27w_c \quad \ldots\ldots\ldots \quad (289)$$

and $$t_c'' = 0.45w_c, \quad \ldots\ldots\ldots \quad (290)$$

with an average of $t = 0.4w_c$ for automobile engines with four-cycle cylinders up to four and *out* of *phase*. For larger engines the limits will increase through

$$t_c''' = 0.50w_c \quad \ldots\ldots\ldots \quad (291)$$

for medium-powered engines to

$$t_c^{\text{IV}} = 0.60w_c \quad \ldots\ldots\ldots \quad (292)$$

for the very large sizes.

128. Fly-wheels.—Single-cylinder gas-engines cannot complete the cycle without fly-wheels or other equivalent rotating parts; multicylinder engines with working impulse every stroke may be able to continue work without them. Thus the first function of the fly-wheel in a gas-engine is to enable a complete cycle to be carried out. There is another function more difficult for the designer to meet; i.e., the maintenance of angular velocity within prescribed limits of variation throughout the cycle. The total revolutions per minute will be kept constant by a governor which really adjusts the *mean effort* to the *mean resistance* over periods of time greater than that for one revolution. Should the actual turning force momentarily increase above the mean resistance, which, neglecting losses, is equal to mean turning force, the angular velocity would momentarily increase, but should the mass rotating be large, the angular-velocity increase would be small.

Change in angular velocity will depend, first, on the difference between actual tangential driving force and the mean; secondly, on the mass and velocity of the revolving parts, and lastly on the length of time the difference in forces continues. It is possible to control these effects in a new engine, as the first and third depend on the uniformity or lack of uniformity of turning effort, which may be made most uniform by properly adjusting inertia and combining cylinders, crank-angles, and cyclic phases. The second factor in the angular-velocity variation is the mass and effective radius of the rotating parts, which may be imagined concentrated in a fly-wheel rim and which rim may be of any size at all, within, of course, certain limits.

In the computation for the weight of rotating parts to limit angular variation it is customary to neglect everything but the rim of the fly-wheel and to concentrate there sufficient metal for the purpose. Anything more, such as cranks, counterweights, hubs, spokes, etc., will exert a helpful effect on steadiness and compensate for errors made in the less exact part of the computation; i.e., that which gives the amount of the variation of turning work above or below the mean in foot-pounds.

129. Weight of Rim for Limiting Velocity Change.—It is possible to exactly calculate the weight of rim necessary to limit change of angular velocity to any desired amount when the difference between the actual energy received in a fraction of a cycle bears a known relation to the mean energy for the whole cycle. This is a problem of pure mechanics.

Let V_1 = maximum velocity of rim in feet per second;
V_2 = minimum " " " " " " "
V = mean " " " " " " "
w_1 = maximum angular velocity of rim in radius per second;
w_2 = minimum " " " " " " " "
w = mean " " " " " " " "
M = mass of rim $= \frac{W \text{ pounds}}{g}$;
ΔE = greatest amount of energy received or lost by the rim in foot-pounds per cycle above the mean, or = greatest area of crank-effort curve above or below the mean to the proper scale to give foot-pounds;

$$K=\frac{V_1-V_2}{V}=\frac{\text{greatest velocity change in rim}}{\text{mean velocity of rim}}=\frac{w_1-w_2}{w}$$

=coefficient of speed variation allowable or the coefficient of unsteadiness;

I = moment of inertia of rim;

then

$$\varDelta E=E_1-E_2=\frac{W}{2g}(V_1^2-V_2^2)=\frac{W}{2g}(V_1+V_2)(V_1-V_2) \quad (293)$$

$$=\frac{W}{2g}(2V)(KV)=\frac{WV^2K}{g} \quad \ldots\ldots\ldots \quad (294)$$

in terms of linear velocity of rim,

$$\text{or} \quad \varDelta E=w^2IK$$

in terms of angular velocity.

The weight of rim necessary for this limit of velocity change to KV feet per second with the intermittently applied energy $\varDelta E$ above or below the mean is therefore

$$W=\frac{\varDelta Eg}{V^2K}\text{ pounds.} \quad \ldots\ldots\ldots \quad (295)$$

129. Energy-fluctuation Coefficients.—Before a wheel can be designed to work within these limits, or before the required weight for the conditions can be found, the quantity $\varDelta E$ in foot-pounds for the particular engine in question must be known. If the diagram of turning efforts can be drawn, this can be found by measuring directly by the planimeter the area involved and applying the proper scale to reduce it to foot-pounds. In general this diagram is not usually available, so that $\varDelta E$ must be calculated from the mean energy by applying thereto a factor determined by such measurements on turning-effort diagram for engines of the same class, operating under the same conditions, as are available. This factor must be so given that $\varDelta E$ can be expressed directly in foot-pounds.

Let Fig. 144 be a turning-effort diagram for a single-cylinder engine whose mean effort is Ab' in pounds and the length of whose crank-path for the cycle of two revolutions is AB. The work done per cycle of one working-stroke is the area $AbcB$, and to the proper scale this represents PLa foot-pounds, and is therefore equal to the indicator card to an approximate scale. The greatest excess energy is (def), which represents ΔE; hence calling the ratio

$$\frac{\text{area } (def)}{\text{area } AbcB} = R_e = \frac{\text{area } (def) \text{ to proper scale}}{\text{area indicator card to proper scale}}, \quad (296)$$

we have

$$\begin{aligned}\Delta E &= R_e \times (\text{area } AbcB \text{ to proper scale}), \text{ foot-pounds,}\\ &= R_e \times (\text{area indicator card to proper scale}), \text{ foot-pounds;}\end{aligned}$$

hence

$$\Delta E = R_e PLa \text{ foot-pounds.} \quad . \quad . \quad . \quad . \quad . \quad (297)$$

Increasing the number of cylinders to four changes, both the mean effort, which is four times that of one cylinder but also changes the relative and absolute value of ΔE, Fig. 145, represents such a turning-effort diagram for four cylinders at 180° phase difference. The base AB as before represents the crank-path for two revolutions and Ab the mean effort for the four cylinders and is four times that for one.

Area $AbcB = 4PLa =$ foot-pounds energy per complete cycle;
Area $AbgC = PLa =$ " " " working-stroke;
Area def represents the greatest energy fluctuation.

It is evident for a multiple-cylinder engine that the ratio of fluctuation may be expressed in two ways, first comparing it to the mean for one working-stroke, as

$$\frac{def}{AbgC} = \frac{\text{foot-pounds represented by } (def)}{PLa} = R_e, \quad . \quad (298)$$

Fig. 144.

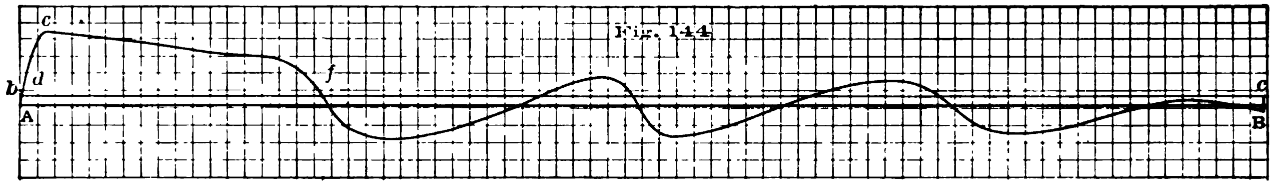
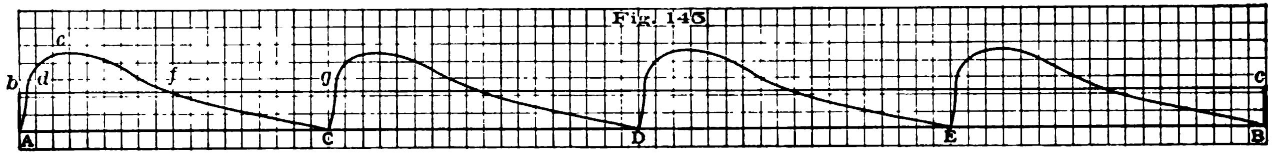


Fig. 145.



or, secondly, comparing it to the mean for the four working-strokes, as

$$\frac{def}{AbcB}=\frac{\text{foot-pounds represented by } (def)}{4PLa}=\frac{R_e}{4}=R_e'. \quad (299)$$

There will then result for η cylinders two ratios: one, R_e, based on PLa, the work per single working-stroke, and $R_e'=\frac{R_e}{\eta}$, based on ηPLa, the work per η working-strokes.

When using these energy-fluctuation coefficients from tables given by different authorities, the designer must be careful to find out which ratio is given. In this work R_e is always given, and therefore

$$\Delta E=R_e PLa \text{ foot-pounds.} \quad . \quad . \quad . \quad . \quad . \quad (300)$$

For the purpose of comparing relative steadiness in reception of energy by the fly-wheel, the other ratio, R_e', is most useful, and this is one reason why the complication of two different ratios is permitted at all. Another reason is that on the turning-effort curve the mean-effort area for the four strokes is most easily measured, because odd phase differences will not make it possible to divide this area into η equal parts, or, in other words, the curve of turning efforts will not repeat itself η times in two revolutions. When thus measured the ratio will be called R_e' and

$$\Delta E=R_e'PLa\eta. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (301)$$

To permit the use of this R_e in the fly-wheel formula for the new engine, it must have been derived from an engine giving a turning-effort curve identical with that expected from the engine in question. Here there will always be of necessity an element of approximation.

Supposing R_e or R_e' to have been properly determined for the new engine, then ΔE can be found:

$$\Delta E=R_e'PLa\eta, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (302)$$

where P = mean effective pressure for the working-stroke or the M.E.P. for the four strokes of a single four-cycle cylinder;

L = length of the working-stroke in feet;

a = area of piston in square inches;

η = number of working-strokes per cycle.

Substituting this, the weight of rim is given by

$$W = \frac{gR_t'PLa\eta}{V^2K} \text{ pounds.} \quad . \quad . \quad . \quad . \quad . \quad (303)$$

If now D_n = mean diameter or twice the radius of gyration of the rim in feet $\frac{\pi D_n N}{60} = V$ = feet per second, giving

$$W = \frac{3600gR_t'PLa\eta}{\pi^2 D_n^2 N^2 K} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (304)$$

$$= 11761\frac{R_t'PLa\eta}{D_n^2 N^2 K}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (305)$$

130. Fly-wheel Formulas.—It is possible here to point out the limitation of a very common fly-wheel formula, which is

$$W' = C_w \frac{La}{D_a^2 N^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (306)$$

For this to be true it is necessary that

$$\frac{R_t'P\eta}{K} = C_w', \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (307)$$

which is equivalent to assuming that all engines have the same mean effective pressure, the same turning-effort curves, and the same coefficient of unsteadiness, or that the coefficient of unsteadiness varies as the product of R_t and mean effective pressure. None of these assumptinos is justified, though for engines precisely alike in all respects the above is true. It may be approximately true for different sizes of engines of the same kind. Sometimes it is convenient to have the fly-wheel weight expressed as a function of indicated horse-power among other factors. This transformation is easily made from formula (305), since

$$\text{I.H.P.} = \frac{PLaN\eta}{2 \times 33000} \quad \ldots\ldots\ldots \quad (308)$$

for η cylinders, each single-acting and working four-cycle, whence

$$PLa\eta = \frac{66000 \times \text{I.H.P.}}{N}. \quad \ldots\ldots \quad (309)$$

Substitution gives

$$W = \frac{11761 \times 66000 \times \text{I.H.P.} \times R_e'}{KD_n{}^2N^3}, \quad \ldots \quad (310)$$

$$= 776226000\frac{R_e'}{KD_n{}^2N^3}\text{I.H.P.}; \quad \ldots\ldots \quad (311)$$

in terms of R.P.M. or partly in terms of linear velocities

$$W = \frac{66000 \times g \times R_e'}{V^2KN}\text{I.H.P.} \quad \ldots\ldots\ldots \quad (312)$$

$$= \frac{2125200R_e'}{V^2KN}\text{I.H.P.} \quad \ldots\ldots\ldots \quad (313)$$

131. Changes in Fluctuation Coefficient with Conditions.—Values of R_e for a number of different conditions, though by no means all, have been found and given below.

Güldner gives

$$R_e = C(0.75 + \zeta), \quad \ldots\ldots\ldots \quad (314)$$

in which $\zeta = 0.30$ for most engines and C has the following values by which R_e can be found.

Two-cycle. c	Four-cycle c	$(0.75+\zeta)$	Two-cycle. R_e	Four-cycle. R_e	Engine.
1.0	.40	1.05	1.05	.42	One-cylinder single-acting
.615	.11	"	.646	.116	" double-acting
.40	.40	"	.42	.42	Two-cylinder tandem-acting
.645	.085	"	.678	.89	" opposed, one crank
.40	.40	"	.42	.42	" " two cranks,
.645	.085	"	.678	.89	set at 360°

These values are a little too general, that is, they do not fit the great variety of conditions met with in practice, nor is the effect of inertia negligible except in the low- and medium-powered stationary engines of low weight of parts and low speeds, producing inertia pressures from 15 to 50 lbs. maximum initial values, as will be seen from an examination of the following values of R_e.

R_e for cylinders 1 to 8 with phase difference $\frac{720^\circ}{\eta}$. Cylinder four-cycle single-acting, on carbureted water-gas with compression of 70 lbs. above atmosphere and maximum pressure 240 lbs. per square inch above atmosphere.

$$R_e = \frac{\Delta E}{PLa}.$$

I. Maximum inertia pressure = 25 lbs. per square inch.

1	2	3	4	5	6	8
1.04	.85	.66	.25	.36	.21	.15

II. Maximum inertia pressure = 50 lbs. per square inch.

1	2	3	4	5	6	8
1.02	.82	.65	.21	.30	.18	.13

III. Maximum inertia pressure = 100 lbs. per square inch.

1	2	3	4	5	6	8
1.01	.79	.67	.64	.33	.18	.14

R_e for cylinders 1 to 8 with phase difference $\frac{720^\circ}{\eta}$. Cylinder four-cycle single-acting, on carbureted water-gas; compression = 65 lbs. per square inch above atmosphere; maximum pressure = 180.

$$R_e = \frac{\Delta E}{PLa}.$$

I. Maximum inertia pressure = 15 lbs. per square inch.

1	2	3	4	5	6	8
.99	.77	.60	.26	.32	.16	.11

II. Maximum inertia pressure = 84 lbs. per square inch.

1	2	3	4	5	6	8
.98	.91	.62	.64	.27	.13	.11

III. Maximum inertia pressure = 207 lbs. per square inch.

1	2	3	4	5	6	8
1.00	1.65	.70	2.02	.29	.56	.09

R_e for cylinders 1 to 8 with phase difference $\frac{720^\circ}{\eta}$. Cylinder four-cycle, single-acting on Taylor producer-gas; compression 90 lbs. per square inch above atmosphere; maximum pressure 183 lbs. per square inch above atmosphere.

$$R_e = \frac{\Delta E}{PLa}.$$

I. Maximum initial inertia pressure = 15 lbs. per square inch.

1	2	3	4	5	6	8
1.00	.94	.73	.28	.37	.19	.17

II. Maximum initial inertia pressure = 80 lbs. per square inch.

1	2	3	4	5	6	8
1.00	1.08	.64	1.09	.33	.22	.13

III. Maximum initial inertia pressure = 207 lbs. per square inch.

1	2	3	4	5	6	8
1.08	2.00	.76	2.90	.18	.16	.14

132. Coefficient of Speed Fluctuation.—Values of the coefficient of unsteadiness or speed-fluctuation coefficient will depend on the service to be performed and the necessary constancy of angular velocity. The most severe requirement is that for engines driving alternators in parallel; the least, that for automobile and boat engines. Automobile engines when in gear with the vehicle have the whole inertia of the vehicle acting as fly-wheel, and require only sufficient fly-wheel proper to help in starting and to keep up the motion when running with gears unmeshed and at low speeds with closed throttle. The weight required is least when four cylinders work on a phase difference of 90°, and is greatest with a single cylinder.

Boat engines act against a practically constant screw-propeller resistance, and require only sufficient fly-wheel to help start or keep up motion when out of gear, and for four or more cylinders this may be nothing at all if the cranks and counterweights are heavy, as these may furnish energy enough for carrying the motion past the end of stroke when the turning force due to terminal pressures is less than the resistance. With early ignition and properly adjusted inertia no fly-wheel need be used.

When the fly-wheel has only such service as the above to perform it is most convenient to give the weight necessary as W_1 pounds per I.H.P. for the different combination of cylinders.

From formula (311) this weight is found by making I.H.P. = 1.

$$W_1 = \frac{776226000 R_e'}{KD^2N^3} \quad . \quad . \quad . \quad . \quad . \quad (315)$$

at a speed N, or energy per I.H.P. is $\frac{MV^2}{2}$, and can be found from

$$W_2 = \frac{2125200 R_e'}{V^2KN} \text{I.H.P.} = \frac{66000 \times g R_e'}{V^2KN} \text{I.H.P.},$$

giving

$$\frac{WV^2}{2g} = \frac{MV^2}{2g} = 33000 \frac{R_e'}{KN} \text{ foot-pounds} \quad . \quad . \quad . \quad (316)$$

per I.H.P. at the speed N, or

$$16.5\frac{R_e'}{KN} \quad \ldots\ldots\ldots \quad (317)$$

foot-tons per I.H.P. at speed N.

For large electric drives, such as alternators in central stations, from one to three foot-tons of fly-wheel energy per I.H.P. are provided, and this is greater than for any other service.

133. **Limiting Diameter of Fly-wheels.**—There is no limit to the diameter of a fly-wheel, except a certain maximum beyond which the centrifugal forces in the rim set up bursting pressures greater than the metal of the rim can resist.

The centrifugal force of the rotating masses acts on the rim just as do internal pressures in a cylinder; but the rim is not free as in that case, since it is fastened by arms to a hub. Neglecting spokes, the force per foot of length tending to part a rim along a diameter is $F=\frac{2wV^2}{gD_w}$ pounds, where w=weight of rim per foot of length. This is resisted by the tension T in the rim on twice the cross-section, a_r, giving

$$T_w=\frac{wV^2}{a_rg} \text{ lbs. per sq. in.} \quad \ldots\ldots \quad (318)$$

$$T_w=0.0000294V^2 \text{ for wrought steel,}$$

$$T_w=0.0000270V^2 \text{ `` cast iron.}$$

From this the safe rim speed at which a wheel may run is, neglecting other stresses,

$$V=\sqrt{\frac{a_rgT}{w}}, \quad \ldots\ldots\ldots \quad (319)$$

and the safe R.P.M. is

$$\frac{\pi DN}{60}=V.$$

$N=\frac{60V}{\pi D^n}$, or the largest diameter for N R.P.M. is

$$D_n=\frac{60V}{\pi N} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (320)$$

This is not correct, because there are other stresses acting, which are:

1°. Initial internal cooling stresses and unknown.
2°. A bending of the rim between arms.
3°. Tension in the arms.
4°. Bending in the arms when speed changes.
5°. Shear in arms and at hub and rim when speed changes.

To simplify the question experiments on bursting of wheels of various forms were made by Prof. Benjamin, and the safe rim velocity determined for each case of solid or built-up wheel, as well as the nature and phase of fracture. It was found that solid-rim wheels were safest and would rupture at speeds of 350 to 400 feet per second. Stresses due to bending are negligible. Jointing the arms at the rim and bracing rims by internal rims have no important effect on strength. Great weakness results from rim joints, which are at best seldom more than one-third as strong as the rim itself.

The Fidelity and Casualty Company, which insures fly-wheels, gives as a safe speed

$$V=1.6\sqrt{\frac{T}{W}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (321)$$

where W = weight of one cubic inch of material in the rim.

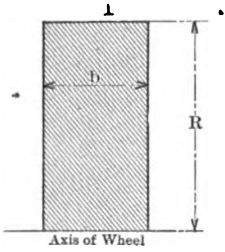
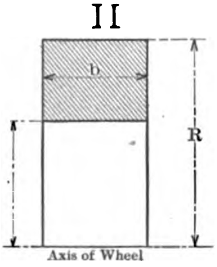
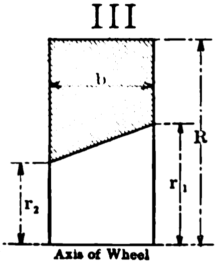
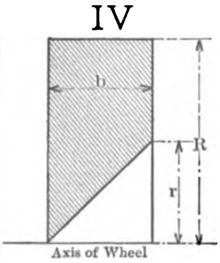
From a knowledge of the energy per horse-power needed and the safe speed of the mean rim the wheel must be designed, that is, its external and internal diameters and width must be determined. This is a matter for which no rule is given, as the result is not dependent on proportions. Hubs and spokes are to be made as in steam-work.

134. **Wheel-rims.**—A convenient table for the energy of wheels of different rims is given by M. E. T. Briggs in "American Machinist."

Then, given the energy, the outside diameter being fixed by the speed, and width of rim being determined by considerations of design, the inside diameter can easily be found; thus

$$r^4 = (R) - \frac{E}{c_1 b_1 N^2}, \quad \dots\dots\dots \quad (322)$$

where E is the total energy and c the constant given in the table.

I 	Dimensions in Feet	Weight in Lbs.	$1426bR^2$
		Energy in Ft.-lbs.	$0.122bN^2R^4$
	Dimensions in Meters	Weight in Kgs.	$22650bR^2$
		Energy in Kg. M.	$6.33bN^2R^4$
II 	Dimensions in Feet	Weight in Lbs.	$1426b(R^2-r^2)$
		Energy in Ft.-lbs.	$0.122bN^2(R^4-r^4)$
	Dimensions in Meters	Weight in Kgs.	$22650b(R^2-r^2)$
		Energy in Kg. M.	$6.33bN^2(R^4-r^4)$
III 	Dimensions in Feet	Weight in Lbs.	$4.77b\left\{3R^2-\left(\frac{r_1^3-r_2^3}{r_1-r_2}\right)\right\}$
		Energy in Ft.-lbs.	$0.0245bN^2\left\{5R^4-\left(\frac{r_1^5-r_2^5}{r_1-r_2}\right)\right\}$
	Dimensions in Meters	Weight in Kgs.	$755.0b\left\{3R^2-\left(\frac{r_1^3-r_2^3}{r_1-r_2}\right)\right\}$
		Energy in Kg. M.	$1.27bN^2\left\{5R^4-\left(\frac{r_1^5-r_2^5}{r_1-r_2}\right)\right\}$
IV 	Dimensions in Feet	Weight in Lbs.	$447b(3R^2-r^2)$
		Energy in Ft.-lbs.	$0.0245bN^2(5R^4-r^4)$
	Dimensions in Meters	Weight in Kgs.	$7550b(3R^2-r^2)$
		Energy in Kg. M.	$1.27bN^2(5R^4-r^4)$

135. Values of Speed-fluctuation Coefficients Practised.—The value of K may be taken as:

K	Class of engine by service;
0.08	Poor regulation and light wheel;
0.04	Shop drives;
0.009	Close regulation shop drives and fair electric drives;
0.005	Good electric drives;
0.003 to 0.0005	Alternators in parallel;
0.002	Average alternators in parallel.

For direct-connected alternators working in parallel the value of K will depend on the dynamo, and specifications will usually be furnished by the dynamo-builder, to be met by the engine-builder or arranged by conference.

When stated, this specification will limit the angular lag or advance of the actual wheel from the position of an absolutely uniformly moving imaginary wheel. The total angular variation is then twice the lag angle allowed on the mean diameter circle, and gives so many feet per second variation permitted from uniformity, from which K is to be determined.

136. Coefficients for Close-regulating Steam-engines.—It is interesting to compare the results given for gas-engines with those of some large steam-engines on electric-light or traction loads. The following table by Mr. A. Marshall Douine is presented for this purpose.

TABLE I.—PARTICULARS REGARDING FLY-WHEELS FOR LIGHTING AND TRACTION STEAM-ENGINES.

Description of Engine.	Description of Generator.	Indicated Horse-power of Engine.	Output of Generator per E.H.P.	Total Weight of Fly-wheel and Rotor.	Foot-tons of Energy at Normal Speed per E.H.P.	Probable Value of Energy-fluctuation Coefficient.	Corresponding Value of Speed fluctuation Coefficient.	Remarks.
Vertical 3-cylinder compound, by Messrs. E. P. Allis & Co. Cylinders 42 in. and 2/60 in. by 60-in. stroke. 75 revolutions per minute.	Three-phase alternator, by Thomson-Houston Co.	4000	2500 K.W. 3400 E.H.P.	Fly-wheel 100 tons Rotor 50 "	4	.035	1/600	As made for Glasgow Corporation Tramway Power Station.
Horizontal 4-cylinder tandem triple-expansion, by the Nurnberg Engineering Co. Cylinders 27½ in., 43½ in., and 2/45½ in. by 63-in. stroke. 72 revolutions per minute.	Three-phase alternator, by Helios E.C. Co.	2800	1700 K.W. 2300 E.H.P.	Fly-wheel generator, 76 tons	3.5	.022	1/2500	As exhibited at Paris, 1900, on lighting load.
Vertical side-by-side compound, by the Nurnberg Engineering Co. Cylinders 30½ in., 49 in., and 71 in. by 43½ in. stroke. 84 revolutions per minute.	Three-phase alternator, 850 K.W., and direct-current generator 750 K.W., by Schuckert Co.	2400	1500 K.W. 2000 E.H.P.	Fly-wheel 40 tons A.C. rotor 30 " C.C. rotor 26 "	3.5	.07	1/500	As exhibited at Paris, one-half on lighting and one-half on power.
Vertical 4-cylinder tandem triple-expansion, by Borsig, Berlin. Cylinders 30 in., 46 in., and 2/52 in. by 48-in. stroke. 90 revolutions. Cranks at 180 in.	Three-phase Siemens and Halske generator	2500	1600 K.W. 2200 E.H.P.	Fly-wheel 39 " Rotor 32 "	3.6	.1	1/380	As exhibited at Paris, 1900, on lighting load.
Vertical cross-compound, by Société Français de Construction Mécanique. Cylinders 32 in. and 68 in. by 48-in. stroke. 75 revolutions per minute.	Three-phase alternator, by Thomson-Houston Co.	1750	1000 K.W. 1450 E.H.P.	Fly-wheel 69 " Rotor 24 "	4.5	.065	1/600	As exhibited at Paris, 1900, on lighting load, but intended ultimately for traction work.
Vertical side-by-side compound, by Société Alsacienne de Construction Mécanique. Cylinders, 33½ in. and 53½ in. by 48-in. stroke. 75 revolutions per minute.	Direct-current generator, by Thomson-Houston Co.	1500	950 K.W. 1250 E.H.P.	Fly-wheel 30 " Rotor 19 "	2.4	.07	1/280	As exhibited at Paris, 1900, on lighting load.
Vertical side-by-side compound, by Messrs. D. Stewart & Co., Ltd. Cylinders, 28 in. and 58 in. by 48-in. stroke. 84 revolutions per minute.	Three-phase alternator, by Oërlikon Co.	1620	1000 K.W. 1340 E.H.P.	Fly-wheel rotor 45 tons	2.8	.07	1/370	As being made for Dublin Corporation Electric Lighting.
Vertical cross-compound, by Messrs. D. Stewart & Co., Ltd. Cylinders 22 in. and 44 in. by 42-in. stroke. 90 revolutions per minute.	Three-phase alternator, by Thomson-Houston Co.	800	500 K.W. 650 E.H.P.	Fly-wheel 33 tons Rotor 10 "	4.5	.065	1/360	As made for Glasgow Corporation Power Station.

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